STEAM POWER AND INTERNAL COMBUSTION ENGINES

BY

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PREFACE TO THE SECOND EDITION

In the second edition, the authors have held to their original purpose—a presentation of fundamental principles of heat-power machinery and a description of the development, construction, and operation of essential equipment of the modern power plant. The importance of engineering methods to determine exact results pertaining to efficiency and performance of various equipment has been stressed.

In the field of internal-combustion engines, the engineering development in the past few years has been phenomenal. Motor ships, locomotives, water plants, and municipal power include a few of the uses in which the oil engine has made great strides. This development has been recognized here by the addition of material on high-speed oil engines. Testing of such engines has been more comprehensively treated.

Likewise in the steam-power field, the use of high-pressure, high-temperature steam has produced striking changes in plant design and operation. The developments resulting therefrom have been carefully outlined, and new material regarding fuel-burning equipment, boiler and furnace design, and boiler-water analysis and treatment in the modern steam plant has been included.

From members of the mechanical engineering staff of Purdue University, especially Professors H. L. Solberg, W. T. Miller, and H. M. Jacklin and Dean A. A. Potter, valuable suggestions have been received which have been of great value in this revision.

Educators from engineering schools over the United States have offered helpful criticisms which have been carefully studied and included when consistent with the purpose of this book. The authors express their gratitude to all who have shown this interest in their book; and especially do they feel that the following engineers have made notable contributions: Professor A. G. Christie, The Johns Hopkins University; Professor J. G. Fairfield, Rensselaer Polytechnic Institute; and Professor D. H. Shenk, Clemson Agricultural College.

The many manufacturers of heat-power equipment have been most generous in responding to requests for photographs, drawings,

and data pertaining to their equipment. Acknowledgment of their courtesy is hereby made.

Appreciation of the courtesy of the American Society of Mechanical Engineers is hereby expressed for their permission to use the Steam Tables by Professor J. H. Keenan.

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PREFACE TO THE FIRST EDITION

This book deals with the fundamental principles underlying heat-power machinery, and it is intended as a textbook for engineering students. It is hoped that engineers in practice will also find valuable information in it.

The subject matter includes descriptions of essential up-to-date mechanical equipment, expositions of theory underlying its construction and operation, and modern methods of adapting it to power units. The historical development has been touched upon where it has been considered effective in increasing the student's interest in heat power.

No attempt has been made to include the electrical equipment of power plants, even though this equipment forms a very important part of such plants. The simpler electrical circuits of automotive engines, however, are described, this being essential to a complete understanding of the operation of these prime movers. Principally as an aid to students who have not studied thermodynamics, there is provided a chapter reviewing the fundamental principles of that subject. To be most effective, a course using this book should be either preceded or accompanied by heat engineering laboratory practice.

The solution of many typical examples has been given to aid in the explanation of methods of making power calculations. Problems follow chapters that contain calculations. For ease of reference and for convenience in assigning parts for study, all subject articles are numbered.

From their colleagues on the instructional staff of Purdue University, the authors have been the recipients of many suggestions that have been of inestimable value in the preparation of this book. Especially do the authors desire to acknowledge their gratitude to Dean A. A. Potter of Purdue University, who has read the manuscript and examined the illustrations. As a result of his careful criticisms, many notable improvements have been made. Professor H. M. Jacklin has given much of his time; and his advice on the material on internal-combustion engines has resulted in a better adaptation of this chapter as a foundation for more advanced courses in automotive engineering and internal-combustion engines. Acknowledgment is also given for general suggestions and assistance to Professor G. A. Young, Head of

the School of Mechanical Engineering, and to Professor C. I.
Hotchkiss, both of Purdue University; to Professor H. E. Deg
Head of the School of Mechanical Engineering, University of Texa
and to Mr. Francis Hodginson and to Mr. S. M. Kintner, of th
Westinghouse Electric and Manufacturing Company.

Credit is hereby given to the authors of the many textbooks whic
have been used as references, and which are listed as such in an Appen
dix. The many manufacturers who have so generously supplie
material are, as far as possible, given credit in the titles of the illustra
tions. It is regretted that because of the large number they canno
be mentioned here.

It is hoped that readers will feel free to communicate to the
authors suggestions, criticisms, and any errors that may be found.

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LAFAYETTE, INDIANA,
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STEAM POWER AND INTERNAL COMBUSTION ENGINES

CHAPTER I

FUNDAMENTALS OF POWER PLANTS

1. Power from Heat.—Fuel, in one form or another, is the source from which nearly all of the mechanical and electrical energy produced by man is derived. The potential heat of a fuel is released and partially transformed into mechanical energy by a system of apparatus which is commonly called a power plant. Fundamentally, a power plant of this nature consists of a combustion chamber where the fuel is burned and its heat is liberated, a medium which absorbs the heat and thereby exerts a pressure, and a machine which receives the heated medium and transforms its energy into mechanical energy. The operation of the plant is, of course, continuous.

The total power generated in the United States during the year 1933 (Bureau of Mines *Report*), on a heat-energy basis was $20,292 \times 10^9$ B.t.u. The percentages of the various contributing sources are given as follows: anthracite coal 7.7, bituminous coal 46.3, American oil 26.9, imported oil 1.1, natural gas 8.7, water power 9.3.

2. Transformation of Heat Energy.—The combustion chamber of a power plant may be a boiler furnace or the cylinder of an engine, and the heat absorbing mediums used are a vapor, such as steam or mercury vapor, or the gases formed by the combustion of a fuel. The machine which actually transforms the heat energy into mechanical energy is called a *prime mover*. Types of prime movers commonly used are the reciprocating steam engine, the steam turbine and the internal-combustion engine. If electrical energy is desired, it is taken from a generator driven by the prime mover.

A steam-power plant consists of a boiler and furnace, a reciprocating steam engine or steam turbine as a prime mover, and the necessary piping, accessories and auxiliary equipment. The steam is generated under pressure in one or more boilers by fuel burned in the accompanying furnace. It then passes through suitable pipes to the prime mover in which expansion takes place. If the steam, after expansion,

is discharged at or above atmospheric pressure, the operation is termed *non-condensing*. If the exhaust steam is discharged into a condenser and reduced to water, in a region where the pressure is less than atmospheric, the operation is said to be *condensing*. The chief advantages of the latter type of operation are increased thermal efficiency and the recovery of the condensed steam. The gain, however, is offset somewhat by the increased cost for the equipment necessary.

The internal-combustion engine plants burn their fuel in the engine cylinder; the heated gases expand by moving a piston and are discharged, near the end of the working stroke, to the atmosphere. The fuel and the air for its combustion are taken into the cylinder at the proper time, and it is ignited just before the start of the working stroke. Exhaust of the expanded gases begins before the end of the stroke. Plants of this type are built principally in small units, as compared with those operating on steam, and higher thermal efficiencies are obtained by them.

Plants using mercury are now in existence, and their performance brings out many advantages. The mercury is vaporized at 35 lb. per square inch gage and a corresponding temperature of 812°F. in an especially constructed boiler. The vapor is expanded in a turbine to 20 in. of mercury vacuum. The exhaust vapor, which is at a temperature of 414°F., is condensed in a water-cooled condenser which serves to generate steam at about 200 lb. per square inch gage. This steam is expanded in an ordinary steam turbine in the usual manner. The gain in thermal efficiency with this type of plant is remarkable, but the gain is greatly offset by the additional cost of the equipment.

3. The Non-condensing Steam Plant.—The simplest form of steam power plant consists, merely, of a relatively inexpensive boiler and furnace and a single cylinder steam engine. The furnace is usually an integral part of the boiler, and the hot gases, formed by the combustion of fuel in the furnace, pass through tubes in the boiler, thence through a smokestack to the atmosphere. Coal is supplied to the furnace, it is burned on the grate, and the ashes are removed by hand. The water is fed to the boiler usually by a steam injector. The steam formed by the absorption of heat through the boiler tubes is piped to the engine where it is expanded and finally exhausted at about atmospheric pressure. The chief accessories used are a pressure gage, safety valve, blow-off valve, all of which are connected with the boiler and furnace, and the governor and lubricator, which are used with the engine. A portable plant of this class usually never exceeds 50 engine hp., and is used principally for portable service, in connection with hoisting and construction equipment. The heat equivalent of the

mechanical energy taken out at the flywheel of the engine seldom exceeds 2 or 3 per cent of the heat supplied to the furnace, in the fuel.

Stationary steam plants of the non-condensing type are usually less wasteful of fuel than those of the portable class. The saving is accomplished by lagging the boiler, piping and engine cylinder with heat-insulating material for reducing the radiation loss. Also, the exhaust steam is frequently used for heating the boiler feedwater or for a steam-heating system, or both. A plant of this class is illustrated in Fig. 1.

The boiler in this figure is of the water-tube type and is enclosed, with the furnace, grate and ash pit, by suitable brick walls. The combined structure is termed the boiler and setting. Coal is fed to the furnace by hand and the ashes are removed by the same means. The combustible gases are distilled from the coal on the furnace grate and are burned, for the most part, in the combustion chamber directly above. Following combustion, the hot gases pass through the three gas passes of the setting and out at the rear, past the damper and through the breeching to the chimney. The flow of the gases is caused by a natural draft produced by the chimney. Steam is taken from the boiler near the front and piped to the main steam header along the wall at the rear. The steam used by the engine is taken from this header and is conducted through the separator, where any moisture is removed, before admitting it to the engine cylinder. The supply of steam is controlled by a governor (not shown in figure) on the flywheel, which in turn controls the speed of the engine. Electrical energy, produced by the connected generator, is wired through the switchboard to service.

The exhaust steam from the engine flows first through an oil separator where the oil admitted for lubrication of the engine valves and cylinder is removed, after which it flows to the feedwater heater and steam-heating system, as shown. The feedwater heater is of the open or induction type, in which the steam gives up its latent heat by intimate contact with the feedwater. Steam for the heating system is taken from the vertical pipe leading to the exhaust head on the roof, and the condensate is returned to the system, through the heater, by gravity. When the demand for exhaust steam is less than the supply, the pressure in the exhaust system is increased and steam is released to the atmosphere through the exhaust head by the opening of the automatic pressure-relief valve.

The condensate from the steam separator is reclaimed and discharged into the feedwater heater by the trap which is automatic in its operation. Cold make-up water is admitted to the system at

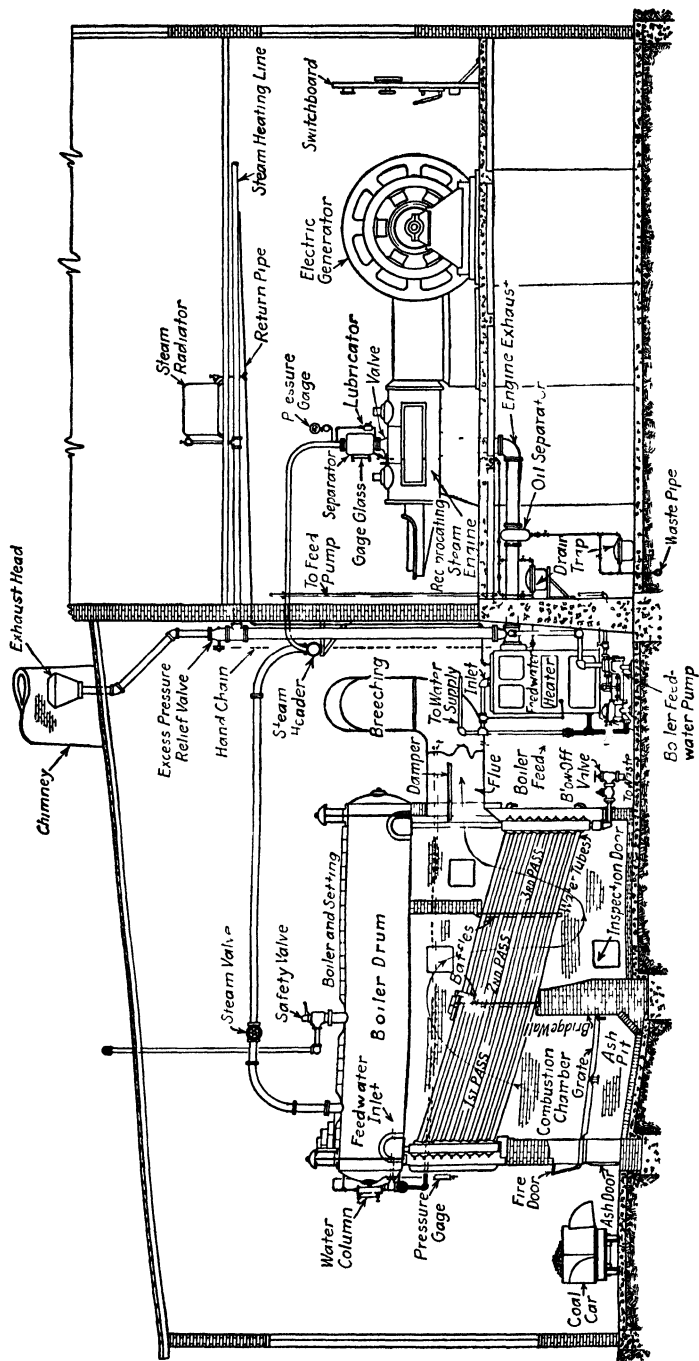



FIG. 1 —Non-condensing steam-power plant.

the top of the heater, as needed, by a valve, as shown, which is operated by a float within the heater. The heated feedwater is taken from the bottom of the heater and pumped to the feed inlet at the front of the boiler drum. The feed pump shown in the figure is of the direct-acting steam type. The steam for operating the pump is taken from the steam line leading to the engine, and the exhaust is directed into the main exhaust line.

A steam plant of the class shown in Fig. 1 generally shows a thermal efficiency of from 6 to 8 per cent. With the addition of a mechanical stoker, superheater, and with forced or induced draft, the thermal efficiency may be raised to as high as 10 to 12 per cent. It is used principally in hotels and small industrial plants where the demand for steam and power is comparatively small, and where a large investment in equipment is unwarranted.

4. The Condensing Steam-power Plant.—The condensing type of plant differs from the non-condensing type in that the exhaust from the prime mover, which may be either a reciprocating steam engine or a turbine, is discharged into a condenser in which the absolute pressure is less than atmospheric. The low-pressure condition is maintained by a rapid condensation of the exhaust steam by either intimate or indirect contact with cold circulating water. The circulating water is passed through the condenser at a rate sufficiently rapid to absorb and carry away the latent heat of the exhaust steam as fast as it is supplied.

The larger plants and central power stations employ condensing operation for their prime movers, and considerable other equipment is usually used for increasing the plant economy. Figure 2 shows a section through a steam-turbine condensing power plant with complete operating equipment.

Coal is delivered from a track hopper by a bucket elevator (neither shown in figure) and a belt conveyor to the overhead bunker. From the bunker it is fed, as required, to the stoker hopper. The chain-grate stoker is mechanically operated and carries the coal into the furnace where it is ignited, burned and reduced to ash which is discharged at the rear, into the ash pit. Air for combustion flows from the front of the furnace through the chain grate. The hot gases pass over the boiler and superheater tubes on traveling through the three vertical gas passes to the rear of the setting, from which they flow through the flue, economizer, induced draft fan (not shown), and breeching to the chimney which is outside the building. The steam piping is so arranged that either saturated or superheated steam may be delivered to the main steam header. In the superheater, additional

heat is absorbed by the steam and its temperature is increased considerably.

The turbine takes its steam from the main steam header. The energy of the steam is given up to the turbine blades, and the spent

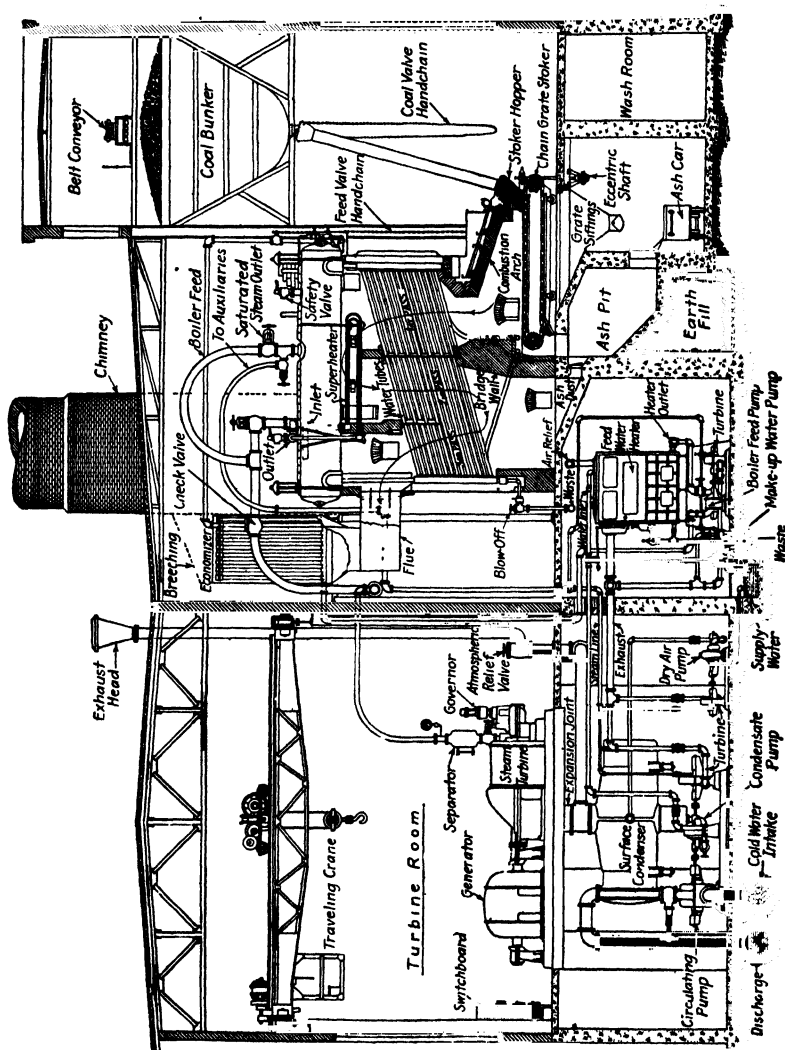



FIG. 2.—Section view of a steam-turbine condensing plant.

or exhaust steam is discharged into a surface condenser in which exists from 24 to 28 in. of mercury vacuum. Condensation of the steam is effected by cold water which is circulated through tubes within the steam chamber of the condenser. An atmospheric relief valve is

provided for automatically releasing the exhaust steam to the atmosphere in case the condenser ceases to function.

The auxiliary apparatus used with the condenser (Fig. 2) consists of a cooling-water circulating pump, a condensate pump for removing the condensed steam, and a dry-air pump for removing air and non-condensable vapors. The condensate is delivered to the boiler feedwater heater where it is mixed with the make-up water. All of the condenser and boiler pumping equipment is driven by saturated steam taken direct from the boiler. The exhaust steam from this equipment is delivered to the heater where it is used for heating the boiler feedwater.

A direct-acting steam pump is used for pumping make-up water from the supply tank into the heater. The heater raises the temperature of the feedwater about 100°F., after which it is delivered by a multiple-stage centrifugal pump to the economizer where the temperature is raised to near that of the boiler steam. From the economizer the feedwater flows directly into the boiler.

A plant of the class just described should give an overall thermal efficiency of 17 per cent or slightly better. In steam turbine plants having extensive auxiliary apparatus, and automatic operation, as high as 28 per cent of heat of the fuel may be converted into electricity under the most economical operating conditions. A section through a medium-sized central power station showing reasonably high operating efficiency is shown in Fig. 3.

The boiler (Fig. 3) is of the water-tube type and the furnace walls are lined with water tubes which connect direct with the boiler. The furnace is fired with pulverized coal, supplied with the air for combustion, through horizontal burners. Raw coal, for pulverizing, is delivered by gravity from the overhead bunkers to the ball-mill pulverizers. With the aid of preheated air, the primary air fans exhaust dry pulverized coal from the mills and deliver it to the several burners. The secondary air, the balance of the air required for combustion, is also preheated, and is delivered to the burners by fans and mixed with the coal as it is blown into the furnace. Considerable heat saving is effected by using both an economizer and an air preheater for absorbing heat that would otherwise be carried away by the flue gases.

The feedwater heating and treating equipment in this plant (Fig. 3) is well arranged to avoid loss of heat and to give high economy. The feedwater is heated in two stages by two separate heaters. The first heater, shown in Fig. 3 as the sixth-stage heater, receives all of the condensate from the plant and heats it to about 170°F. with steam bled from the sixth stage of the turbine (which has nine stages in all).

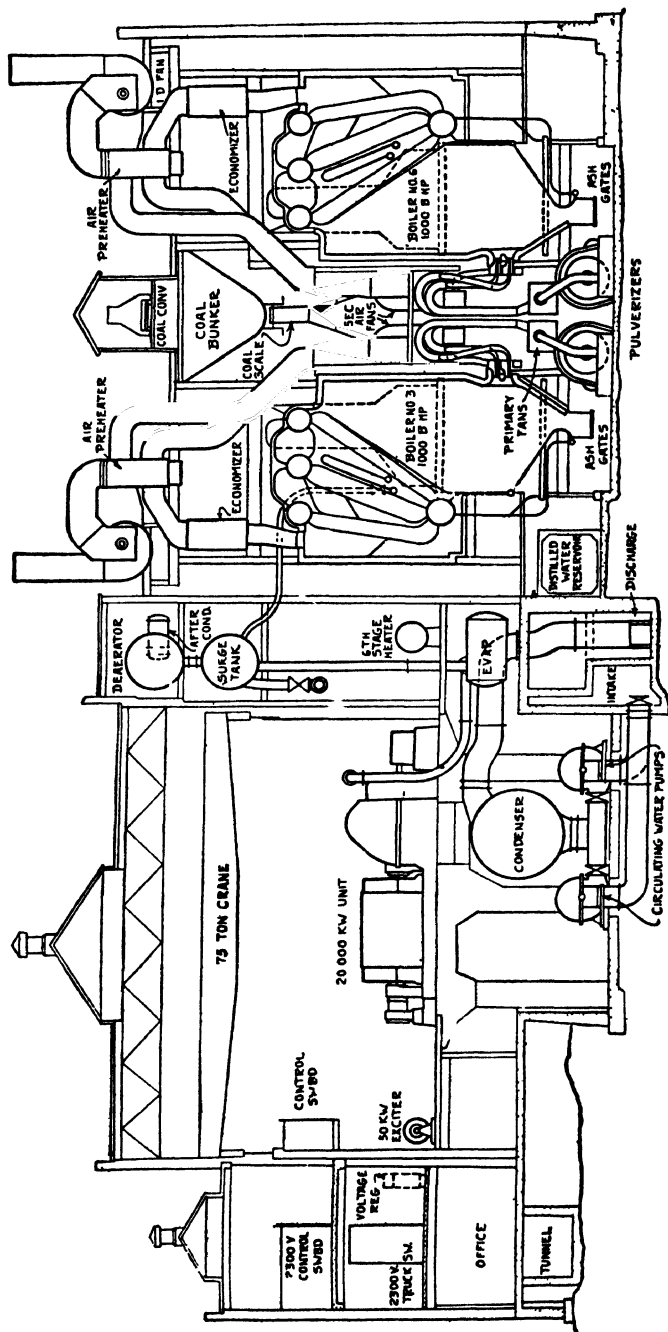



Fig. 3.—Large turbine condensing plant. (*Courtesy of Power.*)

Raw make-up water, which averages about 4 per cent of the water used, is distilled in the evaporators by high-temperature steam bled from the third stage of the turbine. The distilled make-up water vapor is condensed in the de-aerating condenser by the cooler feedwater from the first- or sixth-stage heater. The air which is driven from the raw water during distillation is removed in this condenser. All of the feedwater is collected in the surge tank and from there it is delivered by feed pumps to the second heater, where its temperature is raised to about 265°F. before going through the economizer to the boiler.

The arrangement and types of equipment used in power plants vary over a wide range, and a study of any one plant gives only a general knowledge of modern practice. Steam plants are built to burn almost any kind of fuel successfully, and it should be remembered that the selection of boiler and combustion equipment is largely dependent on the type and grade of fuel to be used. The present tendency in large central stations is toward higher pressures and temperatures and greater rates of burning fuel. Plants are now in operation which generate steam at 1,400 lb. per square inch and 750°F., and the limit of such increase is not yet determined.

5. Internal-combustion Engine Power Plant.—Plants, including vehicles and portable equipment, using internal-combustion engines as prime movers are exceedingly great in number. The types of service vary over a wide range, but it can be generally stated that internal-combustion engines are available for practically all types of service requiring less than 6,000 b.hp. in a single unit. Power plants of this type utilize the cylinders of the prime mover for the combustion chamber, are built in units having one or more working cylinders and operate on the Otto, Diesel or mixed cycle. In most cases they require an elaborate lubrication and cylinder cooling system. The auxiliary apparatus for the smaller plants are usually built integral with and driven from the main shaft of the engine.

Theoretically, internal-combustion engines operating on the Otto cycle effect the burning of the fuel at constant volume. This is accomplished by compressing a mixture of fuel and air during one stroke of the engine and igniting this mixture near the end of the stroke by an electric spark, causing an explosion. The remaining strokes, which may be one or three, provide for expansion of the heated gases, exhaust and intake of a fresh supply of fuel and air.

Engines employing the Otto cycle generally use light oil or gas as fuel. Gasoline engines are of this type and are built in comparatively small sizes for driving automotive vehicles and airplanes and for light

marine service; in fact, practically any kind of service requiring a small amount of power and where intermittent operation is desired.

Internal-combustion engine plants which operate on any of the available gas fuels are built for stationary service and are used principally where there is a plentiful supply of fuel available at low cost. The sizes of these engines range from 10 to about 3,000 b.hp., which is greatly in excess of those using gasoline. The largest sizes are generally double-acting and use blast-furnace gas as fuel.

The Diesel cycle effects the burning of the fuel at constant pressure, and ignition results from a high temperature of air that is compressed

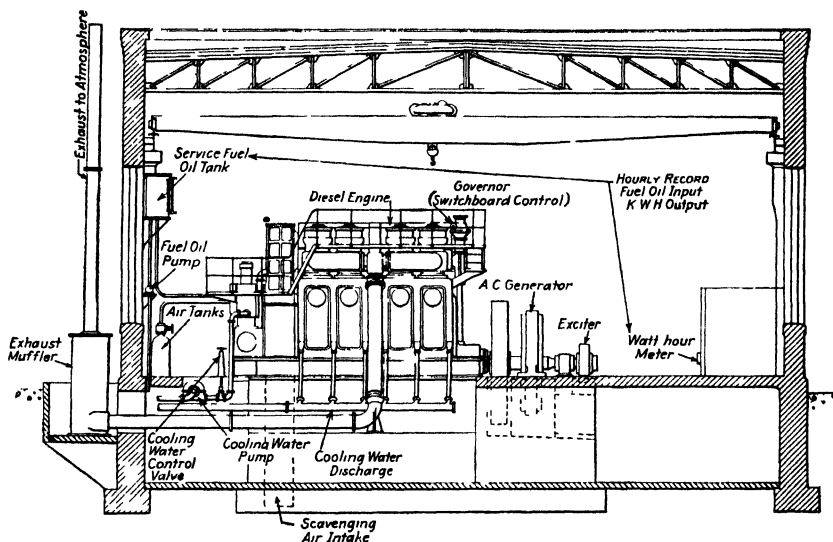



FIG. 4.—Diesel engine power plant. (*Courtesy of Busch-Sulzer Brothers Diesel Engine Company.*)

by the engine piston. Fuel is injected into the cylinder, and silent burning occurs during approximately the first 10 per cent of the stroke, following compression. Diesel engines are built to operate using either two or four strokes to complete the cycle of operation (air intake, compression, fuel injection and burning, expansion and exhaust). The fuels used are generally of petroleum origin ranging from the heavier crude oils to the poorer grades of kerosene, though coal tar and vegetable oils are used to a limited extent.

Oil engines of 15,000-b.hp. capacity have been built, but, in the United States stationary engines of only 10,000-b.hp. capacity, or less, are manufactured. They have a wide range of use, including the driving of air compressors, locomotives, boats, pumps, and electric generating equipment. In electric power stations usually more than

one unit or engine is employed. A section through a typical plant of this type is shown in Fig. 4. The figure illustrates the general location and gives names of those auxiliaries which are separate from the engine.

A Diesel-equipped electric power station of 20,000-kw. capacity would contain about five generating units, equally divided. The first cost of a Diesel plant exceeds that of a steam plant of the same size, but operating and maintenance charges are considerably less. The overall economy of any plant is determined largely by local factors, and it frequently happens that small central stations and isolated generating plants of this type are the best investment.

CHAPTER II

PRINCIPLES OF THERMODYNAMICS

6. General.—Thermodynamics is the science covering changes of energy. It includes a wide field, but in a study of mechanical power, thermodynamics can be limited to a study of heat energy and its transformations. A body is said to possess energy when it is capable of doing work against an external resistance.

The effect of an increase of the stored energy upon a body is explained by the molecular theory of matter. This considers that every substance is made up of minute particles, called molecules, which are in continuous vibratory motion. The relative speed and length of travel of each molecule depend on the temperature, or state of the body, that is, whether gaseous, liquid or solid. As energy is added to a body, either the molecular velocity or the length of the path followed in the molecular motion is increased. This energy, which results in a change in the state of a body, or in a change in the molecular velocity, is known as the internal energy, or intrinsic energy, and may be designated by the symbol U, or u.

The term heat (Q or q) designates energy in flow from one body to another of lower temperature by radiation, conduction and convection. The transfer of heat from the sun to the earth is an example of radiation, and it should be noted that the intervening medium is not heated. The flow of heat through the walls of an internal-combustion engine cylinder may be taken as an example of conduction. The inner surface of the wall is at a much higher temperature than the outer surface, and, as metals are good conductors of heat, there results a flow of heat. Convection refers to the transfer of heat by some carrying fluid, for example, the transfer of heat energy to boiler heat-absorbing surface from the fuel on the grate. The gas formed during combustion is the medium which carries the heat energy to those parts of the boiler which are not exposed to the fuel bed.

Heat (Q or q) and work (W) are not point functions; they do not exist at a point as does internal energy (U or u). Heat and work are developed between the points on the path considered. They are energies flowing from one body or system to another, heat by transfer of energy by temperature difference, and work by transfer of energy

by mechanical force. Internal energy is the stored energy in a body or system in the kinetic and potential energies of the atoms and molecules.

The *units of heat* are the calorie and the British thermal unit. The amount of heat required to raise 1 kg. of water from 17 to 18°C. is called the kilogram-calorie or large calorie. The *British thermal unit* is the amount of heat required to raise 1 lb. of water from 63 to 64°F. The unit commonly used is the mean B.t.u., which is taken as $\frac{1}{180}$ of the heat required to raise 1 lb. of water from 32 to 212°F.

Temperature is a measure of the intensity of the heat of a substance; a hot body has a high temperature while a cold body has a low temperature. The most common method of determining temperature is by the use of a mercury thermometer. For high temperatures electric pyrometers are used. In American engineering practice the Fahrenheit scale for temperature measurement is generally used, while in scientific work the Centigrade scale is universal.

7. Basic Laws of Energy.—The laws of thermodynamics are fundamental facts, unlike mathematical laws that can be logically proved. Most thermodynamical proofs are based on the assumption that the basic laws are true.

The *first law of thermodynamics* is the law of conservation of energy and may be stated generally as follows: Energy cannot be created or destroyed, but it may be changed from one form to another. In reference to heat and mechanical energy, they are indestructible and mutually convertible. This conversion is always effected with the production of 777.6 (taken as 778) foot-pounds of work for each B.t.u. expended, or *vice versa*. Hence, if Q represents the heat converted into work, W,

$$W = JQ$$

or

$$Q = AW$$

in which

$$J = 778$$

and

$$A = 1 \div 778$$

While the first law of thermodynamics states the principle that heat energy can be converted into work, the *second law*, sometimes called the *law of degradation of energy*, gives the restrictions or limitations that influence this conversion. To be available for conversion into mechanical energy, it must be possible for the heat to flow from a body of high temperature, through a heat engine, to a cold body.

The heat engine transforms only a part of the heat energy supplied into mechanical energy. The unconverted portion, which is rejected at the low temperature, is unavailable as far as any additional transformation in the same engine is concerned. The second law may, therefore, be considered to infer that in the transformation of heat into mechanical energy only a part of the heat supplied can be so converted into work; and that the unconverted portion is made unavailable as a source of additional mechanical energy. This principle is also expressed as follows: Every natural process is accompanied by a certain degradation of energy.

The relation between heat supplied, the mechanical energy produced, and the heat rejected is expressed by the following:

$$Q_1 = AW + Q_2$$

or

$$AW = Q_1 - Q_2$$

in which

Q_1 = heat supplied at the high temperature, B.t.u.
Q_2 = heat rejected at the low temperature, B.t.u.
W = work done, ft.-lb.

The *fundamental energy equation*, which deals with changes in energy in a substance, —gas, liquid, or solid,—considers that if the substance receives a definite quantity of heat from an external source, this heat may be distributed in the receiving medium in various forms of energy. Heat absorption may result in an increase in temperature, indicating that the kinetic energy of the molecules has been increased. This is known as *sensible heat*. Heat may cause a rearrangement of the molecular structure without any increase of temperature; for example, the change of a liquid into a vapor. This is known as *latent heat* or potential energy. When heat is added, accompanied by a volume change, work is done against an external pressure, during the absorption process. The heat energy necessary for this work must be supplied from the heat energy absorbed. The fundamental energy equation is as follows:

$$JdQ = JdU + dW \qquad \text{ft.-lb.}$$

or

$$dQ = dU + AdW \qquad \text{B.t.u.} \qquad (1)$$

in which

dQ = quantity of heat added to or taken from the medium.
dU = change in internal energy both sensible and latent.
dW = external work done during the heat change.

In the energy equation, dQ is considered positive if heat is added **and negative if heat is rejected. Likewise, dU is positive if the internal energy is increased; dW is positive if work is performed by the medium, such as during expansion, and negative if work is done on the medium, as in compression.**

The external work may be expressed as follows:

$$dW = PdV$$

Combining this with Eq. (1),

$$dQ = dU + APdV \tag{2}$$

8. Behavior of Gases.—Certain gases were at one time considered to be permanent; that is, it was thought impossible to liquefy such a gas as hydrogen. It is now known that all gases can be changed to a liquid state, at very low temperatures and suitable high pressures. All permanent or so-called perfect gases are really superheated vapors at a temperature far above their temperature of saturation, and they obey very closely the following laws of ideal gases.

Boyle's Law states that, if the temperature be kept constant during a change of state of a given weight of gas, the pressure will vary inversely as the volume; that is,

$$\frac{P_1}{P_2} = \frac{V_2}{V_1}$$

or

$$P_1V_1 = P_2V_2 = C \tag{3}$$

where the subscripts 1 and 2 refer to different states of the gas, at constant temperature. Such a change of state, at constant temperature, is known as an *isothermal change.* On the PV plane, an isothermal curve, $PV = C$, is represented by an equilateral hyperbola with the zero pressure and zero volume lines as asymptotes.

Charles' Law states that if, during a change of state, the volume of a gas is kept constant, the change in pressure is directly proportional to the change in temperature. Also, if the pressure is kept constant, the change in volume is directly proportional to the change in temperature. Experiments have shown that when the pressure remains constant, the volume of a given weight of a gas changes $1/492$ of its value at 32°F. for every degree Fahrenheit change in temperature. Therefore, assuming that a given weight of gas at 32°F. has a volume of 1 cu. ft., and that the pressure remains constant as heat is added to or taken away from the gas, the changes in volume at various temperatures may be shown as follows:

Temperatures, °F.	Volume
524 = 32 + 492	$1 + \frac{492}{492} \times 1 = 2$
32	1
0	$1 - \frac{32}{492} = \frac{460}{492}$
−460 = 32 − 492	$1 - \frac{492}{492} = 0$

If these points are plotted on a temperature-volume diagram, they will be found to lie along a straight line that intersects the temperature axis at −460°F. This would indicate that, with constant pressure, the volume of a gas becomes zero at a temperature of −460°F.; and, with constant volume, the pressure of a gas becomes zero at the temperature −460°F. According to the molecular theory the pressure exerted by a gas is due to the bombardment of the molecules on the enclosing surface. At a point of zero pressure, there is, therefore, no molecular motion. This point of zero molecular motion is called the *absolute zero* of temperature, and temperatures measured from it are known as *absolute temperatures.* Absolute temperatures may use either the Fahrenheit or Centigrade scale.

$$T = t + 273 \text{ on Centigrade scale.}$$
$$T = t + 460 \text{ on Fahrenheit scale.}$$

where

T = absolute temperature.

t = degrees Fahrenheit or degrees Centigrade.

Charles' Law may be stated as follows:

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \qquad \text{(pressure constant)}$$

or

$$\frac{P_1}{P_2} = \frac{T_1}{T_2} \qquad \text{(volume constant)}$$

The two laws, Boyle's and Charles', may be considered together and combined into one statement that will satisfy both.

$$\text{Boyle's Law } PV = C \qquad (T \text{ constant})$$
$$\text{Charles' Law } \frac{P}{T} = C \qquad (V \text{ constant})$$

or

$$\frac{V}{T} = C \qquad (P \text{ constant})$$

Let the initial and final conditions of a given weight of gas be P_1, V_1, and T_1, and P_2, V_2, and T_2, respectively. Starting with the gas at the initial conditions, and changing the volume from V_1 to V_2, at con-

stant temperature, brings the pressure to P', an intermediate pressure. By Boyle's Law, the following relation is true.

$$P_1V_1 = P'V_2 \qquad \text{(with } T_1 \text{ constant)}$$

from which

$$P' = \frac{(P_1V_1)}{V_2}$$

The next step is to change to the final conditions of P_2 and T_2, with the volume constant at V_2. Following the law of Charles:

$$\frac{P'}{T_1} = \frac{P_2}{T_2} \qquad \text{(with } V_2 \text{ constant)}$$

or

$$P_2 = \frac{P'T_2}{T_1}$$

Substituting the value of P' derived from the first step:

$$P_2 = \left(\frac{(P_1V_1)}{V_2}\right) \times \frac{T_2}{T_1}$$

which may be transposed to the following form:

$$\frac{P_2V_2}{T_2} = \frac{P_1V_1}{T_1} = \text{a constant} \tag{4}$$

The constant is denoted by the symbol R and is known as the *gas constant*. It represents the foot-pounds of work done by 1 lb. of the gas for each degree Fahrenheit change in absolute temperature. Thus,

$$Pv = RT \tag{5}$$

and

$$PV = wRT \tag{6}$$

where

P = absolute pressure, lb. per square foot.
v = volume, cu. ft. per pound.
V = volume of w lb. of gas, cu. ft.
T = absolute temperature, °F.
w = weight of gas, lb.
R = gas constant.

The value of R can be determined if the values of P_1, V_1, and T_1 at any definite state are known. For example, the density of air is 0.0807 lb. per cubic foot when the pressure is 14.7 lb. per square inch absolute and the temperature is 32°F. Then from Eq. (5)

$$R = \frac{PV}{wT} = \frac{14.7 \times 144 \times 1}{0.0807 \times 492} \quad 53.35$$

This is the value of R for air, and similar values can be determined for other gases if the specific volume or density is known.

Example 2-1.—Given 3 lb. of air in a rigid container, at an absolute pressure of 25 lb. per square inch, and a temperature of 100°F. What is the volume of the container? If the temperature is increased to 180°F., what will be the resulting pressure?

Solution.—From $PV = wRT$

$$V = \frac{3 \times 53.35 \times 560}{25 \times 144} = 24.9 \text{ cu. ft.}$$

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \qquad (V_1 = V_2)$$

$$\frac{25}{560} = \frac{P_2}{640}$$

$$P_2 = {}^{640}\!/_{560} \times 25 = 28.5 \text{ lb. per square inch.}$$

TABLE 2-1.—PROPERTIES OF GASES

Gas	Molecular symbol	Weight and Volume p = 14.7 lb. abs., t = 32°F.		Mean specific heats	
		Weight per cu. ft., lb.	Volume per lb., cu. ft.	At constant volume, c_v	At constant pressure, c_p
Hydrogen	H_2	0.0056	178.30	2.420	3.408
Nitrogen	N_2	0.0781	12.81	0.173	0.2439
Carbon monoxide	CO	0.0781	12.81	0.172	0.2429
Carbon dioxide	CO_2	0.1228	8.14	0.158	0.203
Methane	CH_4	0.0450	22.20	0.450	0.573
Acetylene	C_2H_2	0.0700	14.28	0.271	0.350
Sulphur dioxide	SO_2	0.1782	5.61	0.123	0.154
Air	....	0.0807	12.39	0.170	0.2385
Oxygen	O_2	0.0892	11.21	0.158	0.220
Helium	He	0.0112	89.29	0.753	1.254

9. Specific Heat of Gases.—If heat is added to a gas, any or all of its conditions of temperature, pressure, or volume will be changed. The *thermal capacity* of a gas may be expressed as the quantity of heat required to effect one unit change in any one of these three conditions. Thermal capacity, as commonly used, is taken as the quantity of heat required to raise the temperature of a unit weight of a gas 1°F.

The *specific heat* of a unit weight of any substance, at a given temperature, is the ratio of the thermal capacity of the substance to the thermal capacity of the same weight of water at standard temperature (60°F.). The thermal capacity of water at 60°F., in English units, is 1 B.t.u. Consequently, the specific heat of any substance at a given temperature is numerically equal to the thermal capacity of 1 lb. of the substance, at the same temperature. Therefore, *specific heat* may be defined as the quantity of the heat which, when added to or taken from 1 lb. of a substance, will change the temperature 1°F. This quantity of heat may change the temperature without doing external work; it may do external work without changing the pressure; or it may change both the pressure and volume while the change in the temperature is being made. There are various values of specific heat, depending on the conditions of pressure and volume during the change. In general, the specific heat (designated by c) may be expressed by the following equation:

$$c = \frac{{}_1Q_2}{w(T_2 - T_1)} \quad \text{B.t.u. per pound per degree Fahrenheit} \qquad (7)$$

in which

${}_1Q_2$ = heat added or subtracted, B.t.u.

w = weight of the substance, lb.

T_1 = temperature before the heat change, °F., abs.

T_2 = temperature after the heat change, °F., abs.

In the case of a gas, the value of the specific heat generally varies, somewhat, through the ordinary temperature range. Owing to this variation, an average value, the *mean specific heat* (see Table 2-1), is used in calculations.

For greatest accuracy in calculating the quantity of heat absorbed or rejected during a change in temperature of a substance, integration may be resorted to. Thus,

$${}_1Q_2 = w\int_{T_1}^{T_2} c\,dT \qquad \text{B.t.u.}$$

For various gases, the variable specific heat is expressed as a function of temperature, and the values most commonly used are those giving the specific heat at constant volume c_v, and the specific heat at constant pressure c_p.

10. Specific Heat of a Gas at Constant Volume.—This value is taken as the quantity of heat which, when added to 1 lb. of the gas at constant volume, will raise the temperature 1°F. The relation

between the quantity of heat and the mean specific heat is given as follows:

$$_1Q_2 = wc_v(T_2 - T_1) \qquad \text{B.t.u.} \tag{8}$$

in which

$_1Q_2$, w, c_v, T_1, and T_2 are as given above.

In a constant volume change of state, there is no work done, as there is no displacement of an external substance. Therefore, the general energy equation [Eq. (1)] becomes, for this case,

$$dQ = dU + 0$$

Thus, in a constant volume change, the change in internal energy is equal to the change in heat energy, expressed in equation form, as follows:

$$U_2 - U_1 = {_1Q_2} = wc_v(T_2 - T_1) \qquad \text{B.t.u.} \tag{9}$$

The internal energy of a gas is a function of temperature alone and does not depend on the intermediate conditions during a change of energy. In other words, for given initial and final temperatures, the change in the internal energy is always the same, regardless of the path followed in going from one state to the other. Therefore, Eq. (9) gives the change in internal energy between the temperatures T_1 and T_2, not only for a constant volume change, but for all other changes of state.

11. Specific Heat of a Gas at Constant Pressure.—The specific heat at constant pressure is the quantity of heat which, when added to a gas, with pressure remaining constant, will raise the temperature by 1°F. In such a change, there is an increase in volume, and, consequently, work is done against external bodies. Since heat is required, not only as internal energy, but also for the external work, the value of c_p will, for any gas, be greater than c_v.

The relation between c_p and c_v may be derived by equating the change in internal energy, between the two stages, by different paths; first, by constant volume, and, second, by constant pressure. Using the same symbols as before,

$$U_2 - U_1 = wc_v(T_2 - T_1) \qquad \text{(constant volume change)}$$
$$U_2 - U_1 = {_1Q_2} - A\int_1^2 PdV \qquad \text{(constant pressure change)}$$

in the above equation,

$${_1Q_2} = wc_p(T_2 - T_1)$$
$$A\int_1^2 PdV = AP(V_2 - V_1)$$
$$= AwR(T_2 - T_1)$$

Then

$$U_2 - U_1 = wc_p(T_2 - T_1) - AwR(T_2 - T_1)$$

But the change in internal energy depends only on the temperatures, which are the same in both conditions.
Therefore,

$$wc_v(T_2 - T_1) = wc_p(T_2 - T_1) - AwR(T_2 - T_1)$$

or

$$c_p - c_v = AR \qquad \text{B.t.u. per pound per degree Fahrenheit} \qquad (10)$$

Thus R, the *gas constant*, represents the foot-pounds of external work done by 1 lb. of the gas when the temperature is increased by 1°F. at constant pressure.

The ratio c_p/c_v is designated by the symbol k, which is constant for a real gas and has a value greater than unity. Combining $c_p/c_v = k$ with Eq. (10) gives

$$c_v = \frac{AR}{k - 1} \qquad \text{B.t.u. per pound per degree Fahrenheit}$$

and

$$c_p = \frac{kAR}{k - 1} \qquad \text{B.t.u. per pound per degree Fahrenheit} \qquad (11)$$

The change in internal energy of a gas may be expressed as

$$dU = wJc_v dT \qquad \text{ft.-lb.}$$

$$U_2 - U_1 = wJc_v(T_2 - T_1)$$

Multiplying and dividing by R, and substituting $1/A$ for J,

$$U_2 - U_1 = \frac{c_v}{AR} wR(T_2 - T_1)$$

but

$$c_v = \frac{AR}{k - 1}$$

and

$$wRT = PV$$

Therefore,

$$U_2 - U_1 = \frac{wR(T_2 - T_1)}{k - 1} = \frac{P_2V_2 - P_1V_1}{k - 1} \qquad \text{ft.-lb.} \qquad (12)$$

Equation (12) gives the change in internal energy for a gas passing from the initial state P_1, V_1, T_1 to the final state P_2, V_2, T_2.

12. Constant Pressure and Constant Volume Heat Changes for a Gas.—During the expansion or compression of a gas, the basic energy

equation holds true; namely, the heat change equals the change of internal energy plus the heat equivalent of the external work done. In practical problems it is usually required to find the change in heat energy, internal energy, or external work done, or one of the conditions of P, V, or T, at a certain state of the gas. The methods used to obtain these results will be outlined for the common types of heat changes which take place in gases.

In a constant pressure change

$$ {}_1Q_2 = wc_p(T_2 - T_1) \quad \text{B.t.u.} \tag{13} $$

$$ A\,{}_1W_2 = AP(V_2 - V_1) = AwR(T_2 - T_1) \quad \text{B.t.u.} \tag{14} $$

$$ U_2 - U_1 = wc_p(T_2 - T_1) - AwR(T_2 - T_1) \quad \text{B.t.u.} \tag{15} $$

or

$$ U_2 - U_1 = wc_v(T_2 - T_1) $$

In a constant volume change

$$ {}_1Q_2 = wc_v(T_2 - T_1) $$

$$ {}_1W_2 = 0 $$

$$ U_2 - U_1 = {}_1Q_2 = wc_v(T_2 - T_1) \quad \text{B.t.u.} \tag{16} $$

13. Isothermal Changes in Gases.—If the temperature of a gas is constant, the characteristic equation $PV = wRT$ becomes $PV =$ constant, and this is the equation of an isothermal change on the P-V diagram. In an isothermal change, the temperature remains constant, and the change in internal energy is zero. Consequently, the heat absorbed is the heat equivalent of the work done ($A\,{}_1W_2$).

$$ dW = PdV $$

$$ {}_1W_2 = \int_{V_1}^{V_2} PdV $$

From

$$ PV = P_1V_1 $$

and

$$ P = \frac{P_1V_1}{V} $$

and substituting

$$ {}_1W_2 = P_1V_1 \int_{V_1}^{V_2} \frac{dV}{V} $$

$$ {}_1W_2 = P_1V_1 \log_e \frac{V_2}{V_1} \quad \text{ft.-lb.} \tag{17} $$

and since

$$P_1V_1 = wRT_1$$

and

$$\frac{V_2}{V_1} = \frac{P_1}{P_2}$$

Eq. (17) may be expressed

$$_1W_2 = wRT \log_e \frac{P_1}{P_2} \qquad \text{ft.-lb.} \tag{18}$$

since

$$_1Q_2 = A\,_1W_2$$

$$_1Q_2 = AwRT \log_e \frac{V_2}{V_1} \qquad \text{B.t.u.} \tag{19}$$

14. Adiabatic Changes in Gases.—In an adiabatic expansion or compression no energy in the form of heat is added to or subtracted from the medium. If a gas is expanded in a perfectly insulated cylinder, the expansion would then be adiabatic. In the energy equation [Eq. (1)] dQ is, therefore, zero, and work is done at the expense of the internal energy.

By combining the energy equation with the characteristic gas equation the pressure-volume relation of an adiabatic change can be obtained, as follows:

$$dQ = dU + APdV$$

for 1 lb.

$$0 = c_v dT + APdV \tag{a}$$

and

$$PV = RT \tag{b}$$

Differentiating (b) and multiplying by A

$$AVdP + APdV = ARdT = (c_p - c_v)dT$$

or

$$AVdP + APdV - (k - 1)c_v dT = 0$$

From (a)

$$c_v dT = -APdV$$

$$AVdP + APdV + (k - 1)APdV = 0$$

$$VdP + kPdV = 0$$

Dividing by PV

$$\frac{dP}{P} + k\frac{dV}{V} = 0$$

and

$$\log P + \log V^k = \log C$$

which can be changed to the form

$$PV^k = C \tag{20}$$

which gives the relation between pressure and volume during any adiabatic expansion or compression.

The relation between the initial and final conditions of P, V, and T, during an adiabatic change, may be determined by using Eq. (20) with the law of perfect gases, Eq. (4).

$$P_1V_1^k = P_2V_2^k \tag{21}$$

Dividing this equation by

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

gives

$$V_1^{(k-1)}T_1 = V_2^{(k-1)}T_2$$

or

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{(k-1)} \tag{22}$$

Similarly, the following relation may be obtained

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\left(\frac{k-1}{k}\right)} \tag{23}$$

The work done during an adiabatic change may be determined by integrating the expression; thus

$$_1W_2 = \int_{V_1}^{V_2} PdV$$

$$PV^k = P_1V_1^k$$

and

$$P = P_1\frac{V_1^k}{V^k}$$

$$_1W_2 = P_1V_1^k\int_{V_1}^{V_2} V^{-k}dV$$

$$= P_1V_1^k \times \frac{V_2^{(1-k)} - V_1^{(1-k)}}{1-k}$$

$$= \frac{P_1V_1^kV_2^{(1-k)} - P_1V_1^kV_1^{(1-k)}}{1-k}$$

In the first term, substituting for $P_1V_1^k$ its equal $P_2V_2^k$ and multiplying the numerator and denominator by -1, the above equation becomes

$$ {}_1W_2 = \frac{P_1V_1 - P_2V_2}{k-1} \quad \text{ft.-lb.} \tag{24}$$

Eliminating the final volume V_2

$$ {}_1W_2 = \frac{P_1V_1}{k-1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}\right]. \quad \text{ft.-lb.} \tag{25}$$

The change in internal energy is equal to the external work but is given the opposite sign.

Example 2-2.—In a Diesel engine cylinder containing 0.35 lb. of air, during the compression stroke, the absolute pressure rises from 14.7 lb. per square inch to 465 lb. per square inch. If the initial temperature is 75°F., what is the final temperature and the work of compression?

Solution.—*a.* From Eq. (23)

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}$$

$$T_2 = 535\left(\frac{465}{14.7}\right)^{\left(\frac{1.4-1}{1.4}\right)} = 1438\text{°F. Abs.}$$

b. For the work of compression Eq. (25), with WRT substituted for P_1V_1, is used

$$ {}_1W_2 = \frac{WRT}{k-1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}\right]$$

$$ {}_1W_2 = \frac{0.35 \times 53.35 \times 535}{1.4 - 1}\left[1 - \left(\frac{465}{14.7}\right)^{\left(\frac{0.4}{1.4}\right)}\right] = 42{,}100 \text{ ft.-lb.}$$

15. General Expression for Pressure and Volume Changes in Gases.—All of the common changes in pressure and volume of gases can be represented by the expression $PV^n = C$, in which n is a general term having a specific value in particular cases. By giving n special values, this expression is made to cover the familiar changes of state, as for instance:

$n = 0$	$PV^0 = C$	constant pressure change.
$n = 1$	$PV^1 = C$	isothermal.
$n = k$	$PV^k = C$	adiabatic.
$n = \infty$	$PV^\infty = C$	constant volume.

General changes of state covered by the expression, $PV^n = C$, are commonly termed *polytropic changes.* The practical use of this type of change lies in the fact that most expansions or compressions of gases in heat engines follow a polytropic curve, in which the value of n

lies between 1 and k. In other words, the common polytropic curve lies between the isothermal and adiabatic on the P-V diagram.

By combining $P_1V_1^n = P_2V_2^n$ with $P_1V_1/T_1 = P_2V_2/T_2$, the following relations are obtained:

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \tag{26}$$

For the external work, the development is similar to that of adiabatic expansion.

$${}_1W_2 = \frac{P_1V_1}{n-1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)}\right] \quad \text{ft.-lb.} \tag{27}$$

The change in internal energy is, from Eqs. (9) and (12), the same in any change of state.

16. Flow of Gas through an Orifice.—The basic law in the study of flow is that energy cannot be created or destroyed but can be changed from one form to another. If a constant weight per second of fluid undergoes changes of energy while in flow, an energy relation between two points in the path of flow can be determined. This is done by equating the sum of the energies at the selected points. Thus, for 1 lb. of fluid:

$$Az_1 + A\frac{V_1^2}{2g} + u_1 + AP_1v_1 + {}_1Q_2 = Az_2 + A\frac{V_2^2}{2g} + u_2 + AP_2v_2 + A({}_1W_2)$$

in which

z = height above datum plane, ft.
V = velocity of flow, ft. per second.
u = internal stored energy, B.t.u.
Pv = flow work, ft.-lb.
${}_1Q_2$ = heat flow (in + or out −) between 1 and 2, B.t.u.
${}_1W_2$ = energy change as work (out + or in −) between 1 and 2, ft.-lb.

Specific cases of flow start with this equation, and a simpler relation is obtained by canceling items of energy which equal zero, and canceling equal items at the two points.

When a perfect gas flows through an orifice or nozzle, with frictionless flow the above equation becomes

$$A\frac{V_1^2}{2g} + u_1 + AP_1v_1 = A\frac{V_2^2}{2g} + u_2 + AP_2v_2$$

By rearrangement and substitution for the change in internal energy (adiabatic flow), the kinetic energy change is obtained. This is summarized in the following equation, based on 1 lb. of gas:

$$\frac{V_2^2 - V_1^2}{2g} = \frac{-(P_2v_2 - P_1v_1)}{k - 1} + P_1v_1 - P_2v_2 \tag{28}$$

in which

V_1 and V_2 = velocities, ft. per second, at entrance and exit.
P_1 and P_2 = absolute pressure, lb. per square foot, at entrance and exit.
v_1 and v_2 = specific volumes, cu. ft. per pound, at entrance and exit.
g = acceleration of gravity (32.17).
k = constant for the gas.

If the initial velocity is zero, which is usually the case, the final velocity in the above expression becomes

$$V_2 = \sqrt{2g\frac{k}{k-1}(P_1v_1 - P_2v_2)} \quad \text{ft. per second} \tag{29}$$

Designating the weight (lb.) of gas discharged per second by w, and the area of the orifice (sq. ft.) by A,

$$wv = AV$$

or

$$wv_2 = AV_2 \tag{30}$$

By solving this equation for V_2, and the relation $P_2v_2^k = P_1v_1^k$ [Eq. (21)] for v_2, and substituting these values in (Eq. 29) and simplifying, the weight of discharge may be expressed as follows:

$$w = A\sqrt{2g\left(\frac{k}{k-1}\right)\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{k}}\right]} \quad \text{lb. per second} \tag{31}$$

By inspection, it will be seen, from Eq. (31), that when $P_2 = P_1$, and when $P_2 = 0$, the the discharge will be zero. Experiments, however, show the latter deduction to be false. Experiments also show that on reducing the pressure P_2 below P_1 (likewise, increasing P_1), the amount of gas discharged increases to a definite maximum point. Further increase in the difference $(P_1 - P_2)$ beyond this point has no effect on increasing the discharge, and it is found that the pressure at the mouth of the orifice, maintains a definite value, called the *critical pressure* (designated by P_0).

To find the relation of the critical pressure P_0 to the initial pressure P_1, it is necessary to find the value of P_2, in Eq. (31), that will give a maximum discharge w. It is evident that w is a maximum when $\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{k}}\right]$ is a maximum. By taking the first derivative of this term, and equating it to zero, the following relation is obtained:

$$\frac{P_2}{P_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{32}$$

Substituting for P_2 the critical pressure symbol P_0, this equation becomes

$$\frac{P_0}{P_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{33}$$

By substituting the value of k (1.4) for air in Eq. (33), and solving, the critical pressure for air is found to be $0.53P_1$. This indicates, that the maximum amount, per second, of air that can be discharged through an orifice is attained when the pressure on the exit side is 0.53 of the initial pressure.

Constants for other gases can be obtained by substituting the respective values of k in Eq. (33) and solving for P_0.

17. Entropy.—A certain property of all substances, called *entropy*, represented by the symbol s or S, is used in theoretical considerations of energy changes and in the solution of practical heat-engine problems. Entropy is an abstract ratio and can be defined for reversible (frictionless) energy changes by stating that for an infinitesimal quantity of energy in transition (heat flow), the quantity of energy in flow divided by the absolute temperature of the medium affected is equal to the infinitesimal change of entropy. Expressed mathematically this definition is

$$ds = \frac{dQ}{T} \quad \text{or} \quad dQ = Tds \tag{34}$$

The Carnot theoretical cycle (see Art. 296) gives the maximum possible efficiency of any heat engine working between a certain source of heat at T_1 and a refrigerator at T_2. This implies an ideal engine working on the reversible Carnot cycle. The thermal efficiency is expressed as

$$e_t = 1 - \frac{T_2}{T_1}$$

For an infinitesimal quantity of heat (dQ) supplied to the Carnot reversible engine the part which would be available for transformation into work is

$$\text{Available energy} = \left(1 - \frac{T_2}{T_1}\right)dQ \quad \text{B.t.u.}$$

The unavailable part is

$$\text{Unavailable energy} = \left(\frac{T_2}{T_1}\right)dQ \quad \text{or} \quad T_2\left(\frac{dQ}{T_1}\right) \quad \text{B.t.u.} \quad (35)$$

Hence it is seen that the infinitesimal change in entropy in a reversible process is a measure of the unavailability of thermal energy for transformation into work. In such a process the product of the refrigerator temperature and of the change in entropy gives the quantity of energy which is unavailable for transformation into work in the ideal engine used.

Entropy changes and not total quantity of entropy can be determined for any medium. Entropy like stored energy is a state, or point, function and depends upon the properties of the medium at the start and end of a process, and not upon the path followed during the process. The total change in entropy of a system is the sum of the entropy changes of the parts of the system.

The total entropy change may be determined by integration. Assuming frictionless change of state,

$$s_2 - s_1 = \int_1^2 \frac{dQ}{T}$$

The entropy change that occurs during the special heat changes, which have already been discussed, can be determined by substituting the proper value of dQ in the above equation.

1. Constant volume change:

$$s_2 - s_1 = wc_v\int_1^2 \frac{dT}{T} = wc_v \log_e\left(\frac{T_2}{T_1}\right) \quad (36)$$

2. Constant pressure change:

$$s_2 - s_1 = wc_p\int_1^2 \frac{dT}{T} = wc_p \log_e\left(\frac{T_2}{T_1}\right) \quad (37)$$

3. Constant temperature change:

$$s_2 - s_1 = \frac{{}_1Q_2}{T} = \frac{AwRT \log_e \frac{V_2}{V_1}}{T} = AwR \log_e\left(\frac{V_2}{V_1}\right) \quad (38)$$

4. Adiabatic change:

$$s_2 - s_1 = \frac{dQ}{T} = 0 \tag{39}$$

18. Enthalpy.—Enthalpy (H or h) is a useful property of a medium and is defined as the sum of the stored internal energy and the flow work. The term PV designates the flow work or the energy that would be required to cause a quantity of medium with volume V to displace its own volume against the pressure P.

$$H = U + APV \qquad \text{B.t.u.}$$

In the study of vapors, the terms *heat content* or *total heat* have been widely used for this property.

If the fundamental energy equation [Eq. (2), Art. 7] is considered for a constant pressure process,

$$dQ = dU + APdV$$

By integration

$$\begin{aligned} {}_1Q_2 &= (U_2 - U_1) + AP(V_2 - V_1) \\ {}_1Q_2 &= (U_2 - U_1) + APV_2 - APV_1 \\ {}_1Q_2 &= (U_2 + APV_2) - (U_1 + APV_1) \\ {}_1Q_2 &= H_2 - H_1 \qquad \text{B.t.u.} \end{aligned} \tag{40}$$

Hence for a constant pressure process, the energy in transition (heat flow) is equal to the change in enthalpy for any medium. Enthalpy is another point function depending only on the properties of the medium at the start and finish of a process and not upon the path followed. The usefulness of enthalpy is particularly evident in steam-generation calculations, as the heat absorption by the water and steam flowing through the steam-generating unit is a constant pressure process, theoretically.

19. Gas Mixtures.—A so-called *compound gas* is a mixture of two or more different gases. Such a mixture may be considered as being in a container. By Dalton's Law, the total pressure exerted upon the walls of the container is equal to the sum of the pressures due to each of the constituent gases. Each constituent gas behaves exactly as if it alone occupied the volume of the container. If the subscripts 1, 2, 3, etc., be used to designate the different gases, then the characteristic equations can be set up:

$$\begin{aligned} P_1V &= W_1R_1T \\ P_2V &= W_2R_2T \\ P_3V &= W_3R_3T, \qquad \text{etc.} \end{aligned}$$

Adding these equations,

$$(P_1 + P_2 + P_3, \cdots)V = PV = (W_1R_1 + W_2R_2 + W_3R_3, \cdots)T$$

For the mixture, the equation is

$$PV = WRT$$

In the above equations,

P = total pressure, lb. per square foot.
V = volume of gas mixture, cu. ft.
P_1, P_2, P_3, etc. = partial pressures of the constituents, lb. per square foot.
W_1, W_2, W_3, etc. = weights of constituents, lb.
W = total weight of mixture, lb.
R_1, R_2, R_3, etc. = gas constants of constituents.
R = gas constant of mixture.
T = absolute temperature of mixture, °F.

From the last two equations the value of R, the gas constant for the mixture, can be derived:

$$R = \frac{w_1}{w}R_1 + \frac{w_2}{w}R_2 + \frac{w_3}{w}R_3, \quad \text{etc.} \tag{41}$$

By considering the heat absorbed by a gas mixture during a constant pressure (or constant volume) process, and equating this to the summation of the heat absorbed by the constituent gases, the value of the specific heat for the mixture can be determined. Thus,

$$c_p = \frac{w_1}{w}c_{p_1} + \frac{w_2}{w}c_{p_2} + \frac{w_3}{w}c_{p_3}, \quad \text{etc.} \tag{42}$$

$$c_v = \frac{w_1}{w}c_{v_1} + \frac{w_2}{w}c_{v_2} + \frac{w_3}{w}c_{v_3}, \quad \text{etc.} \tag{43}$$

A useful unit of volume in the solution of volumetric relations of a gas mixture, or of different gases is the *mol* or *pound-mol*. The mol is a volume of gas, whose weight expressed in pounds is numerically equal to the molecular weight of the gas. This is true for a gas mixture as well as for a single gas. Under the same conditions of pressure and temperature, the volume of a mol is constant for all gases. This is based on *Avogadro's Law*, which states that, under the same conditions of pressure and temperature, equal volumes of any different gases contain the same number of molecules. The volume of 1 mol of any gas at a pressure of 14.7 lb. per square inch, absolute, and 32°F. is

358.7 cu. ft. Applying the characteristic perfect gas law to a mol of a specific gas such as oxygen,

$$V = \frac{MRT}{P} = \frac{32 \times 48.24 \times 492}{14.7 \times 144} = 358.7 \text{ cu. ft. per mol.}$$

Since, with constant conditions of pressure and temperature, the volume of a mol is constant for all gases or gas mixtures, the characteristic equation of a perfect gas may be expressed on a mol basis as

$$MR = \frac{PV}{T} = 1{,}544 = R_u \tag{44}$$

in which

M = weight of gas (equal to molecular weight), lb.
R = gas constant.
V = volume of 1 mol at P and T, cu. ft.
P = pressure, lb. per square foot absolute.
T = absolute temperature, °F.

The constant 1,544, is obtained by substituting values of P and T with the proper value of the volume of 1 mol. It is termed the *universal gas constant* and is constant for all gases.

20. Vapors.—There is an intermediate condition when a substance is changing from a liquid to a gaseous state, and in this condition the laws of gases do not hold. Substances in this state are known as *vapors,* common examples of which are steam and ammonia. When vapors are heated to a temperature considerably above their vaporizing temperature, they follow, approximately, the perfect gas laws, and the higher the temperature to which they are heated the more accurately they follow these laws.

Steam is the vapor most widely used in heat engines, and it will be the only one discussed. In general, other vapors have properties similar to those of steam. When water is heated, its temperature will rise at a uniform rate (c_p at constant pressure for water being approximately 1.0) until a certain temperature is reached. This temperature is called the *boiling* point or *saturation temperature* and depends on the absolute pressure exerted on the water. For example, at an absolute pressure of 14.7 lb. per sq. in. the saturation temperature is 212°F., and at 100 lb. per square inch it is 328°F.

An important difference between gases and saturated vapors is: at a certain pressure, a gas can have different temperatures by making corresponding changes in volume, while a saturated vapor at a certain pressure can have only one temperature.

When sufficient heat has been added, all of the water will be changed to vapor, and the steam is said, then, to be dry and saturated. This change of state is called *vaporization*. If this vapor is heated further, at constant pressure, its temperature will again increase, and it will become superheated steam.

21. Steam.—During the formation of steam in a boiler, the water boils rather violently, and some water, in the form of a fine mist, is entrained in the emitted vapor. The vapor in this condition is termed *wet saturated steam*. If all liquid is eliminated, the vapor is called *dry saturated steam*. In either case the temperature is the saturation temperature corresponding to the pressure of the steam. If heat is taken from saturated vapor, with no change in pressure or temperature, a part of the vapor will be condensed to liquid. Saturated vapor may be defined as a vapor so near the liquefying point that the removal of heat will result in partial condensation.

For saturated steam, the quality designated by the symbol x is defined as the ratio of the weight of dry saturated vapor to the total weight of the mixture of vapor and entrained liquid. The quality or dryness may be used as a fraction or as a percentage. Thus, if x is 0.9 or 90 per cent, it means that, in each pound of saturated vapor, 0.9 lb. is dry steam and 0.1 lb. is water.

In measuring the quantity of heat in water and steam, the steam tables customarily start with 1 lb. of water at 32°F. This is also the datum temperature from which measurements of entropy and internal energy are taken.

22. Enthalpy of Water and Steam.—The term *saturated*, when applied to water, indicates that the water temperature is at that of saturation (boiling). Any additional heat flow to the water will start steam formation, if the pressure and temperature remain constant. The *enthalpy of the saturated liquid* (total heat of the saturated liquid) is designated by the symbol h_f and, for any given temperature, is equal to the change in enthalpy for 1 lb. of water from the datum temperature, 32°F., to the given temperature. Hence,

$$h_f = u_f + APv_f \qquad \text{B.t.u. per pound}$$

$$h_f = \int_{32^\circ\text{F.}}^{t} c_p dt \qquad \text{B.t.u. per pound}$$

$$h_f = t - 32^\circ\text{F.} \qquad \text{B.t.u. per pound}$$

In the above equations t designates the saturation temperature in degrees Fahrenheit. The last equation is approximate and should not be used for temperatures above 300°F. For accurate values of h_f, tables of properties of saturated water and steam should be used.

If 1 lb. of water at saturation temperature absorbs heat with no change in pressure or temperature, steam will form. If this heat absorption continues until vaporization of the pound of steam is complete, it will contain no liquid, and is then called *dry saturated steam.* The increase in enthalpy during the vaporization process is named the *enthalpy of evaporation* (total heat of evaporation) and is designated by the symbol h_{fg}.

$$h_{fg} = u_{fg} + APv_{fg} \qquad \text{B.t.u. per pound}$$

The *enthalpy (total heat) of dry saturated steam* designated by the symbol h_g is the sum of the enthalpies of both the saturated water and evaporation. The *enthalpy (total heat) of wet saturated steam* is the sum of the enthalpy of the liquid and that part of the enthalpy of evaporation which has been absorbed. The *quality* gives the percentage of dry steam and likewise the percentage of absorption of the enthalpy of evaporation.

For wet steam:

$$h = h_f + xh_{fg} \qquad \text{B.t.u. per pound}$$

For dry steam:

$$h = h_g = h_f + h_{fg} \qquad \text{B.t.u. per pound}$$

If dry saturated steam not in contact with water is heated at constant pressure, the temperature will rise above that of saturation. This produces superheated steam. The *enthalpy (total heat) of superheated steam* is equal to the enthalpy of dry saturated steam plus the increase in enthalpy during the superheating process.

For superheated steam:

$$h = h_g + \int_t^{t_s} c_p dt \qquad \text{B.t.u. per pound}$$

In the above quations, the following symbols are used:

h_f = enthalpy of liquid, B.t.u. per pound.
h_{fg} = enthalpy of evaporation, B.t.u. per pound.
h_g = enthalpy of dry saturated steam, B.t.u. per pound.
h = enthalpy, B.t.u. per pound.
x = quality or dryness, proportion by weight.
t = saturation temperature, °F.
t_s = superheated steam temperature, °F.
c_p = specific heat, of water (approximately 1.0);
of superheated steam (0.45 − 0.75).

23. Temperature-entropy Diagram.—A better understanding of the component parts making up the enthalpy of steam may be had by a study of Fig. 5. On this diagram the ordinate is absolute temperature, and the abscissa is in units of entropy, and the areas represent heat quantities in B.t.u. per pound. The condition of water at 32°F. is represented on the diagram by point A; at this point the total heat and entropy are zero. Assuming that 1 lb. of water is kept under a constant absolute pressure of 150 lb. per square inch, and heat is added, the temperature will increase up to 358.4°F., and the entropy will also increase. This heat is added along the line AB and the area

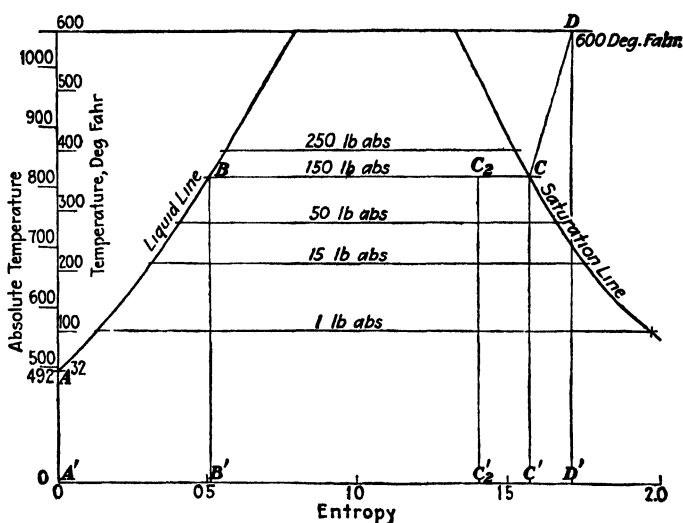



FIG. 5.—Temperature-entropy diagram for water and steam.

$A'ABB'$ represents the enthalpy of the liquid h_f. A further addition of heat, at constant pressure, causes vaporization at constant temperature and a considerable increase in entropy. The state of incomplete vaporization or wet steam is represented by C_2, and xh_{fg} is represented by the area $B'BC_2C_2'$. From C the addition of heat causes an increase both in temperature and in entropy. If the heating is assumed to stop at D (t_s = 600°F.), the heat of superheat $c_p(t_s - t)$ is represented by the area $C'CDD'$.

The path $ABCD$ represents the path followed on the temperature-entropy (TS) plane when heat is added to water and steam at constant pressure. The area under the curve at any point represents the enthalpy of the water or steam for the condition represented by that particular point.

Referring to Fig. 5, it may be seen that, in a constant pressure process, as the enthalpy of steam increases the entropy also increases. In all natural changes there is a tendency to increase the entropy, and any process that results in a decrease in entropy is brought about by some external influence. The entropy of a system depends upon the mass; the entropy of W lb. of steam equals W times the entropy of 1 lb. Entropy, like heat, is relative, and changes of entropy, instead of total entropies, are measured. Values of total entropies given in steam tables are changes of entropy from the condition of 1 lb. of water at 32°F. It is a function of state similar to internal energy in that changes of entropy depend on the initial and final conditions only and not on the path followed during the change.

In a reversible adiabatic change, there is no heat added or withdrawn from the medium, and, therefore, $dQ = 0$. From this it is seen that for such a change $ds = dQ/T = 0$. On the T-S steam diagram, adiabatic changes are represented by vertical lines, and isothermal changes by horizontal lines. The foregoing general statements are not limited to steam alone, but apply to other substances also.

Referring again to Fig. 5, it may be seen that the total entropy of superheated steam is made up of three parts: the entropy of liquid $A'B'$, the entropy of vaporization $B'C'$, and the entropy of superheat $C'D'$. The symbols used for these entropies and the method of calculating the values are as follows:

$$\text{Entropy of liquid} = s_f = \int_{492}^{T} \frac{c'_p dT}{T}$$

$$\text{Entropy of vaporization} = s_{fg} = \frac{h_{fg}}{T}$$

$$\text{Entropy of superheat} = s = \int_{T}^{T_s} \frac{c_p dT}{T}$$

in which

c'_p = specific heat of water.
c_p = specific heat of superheated steam.
T_s = absolute temperature of superheated steam, °F.
T = absolute temperature of saturated steam, °F.
h_{fg} = enthalpy of evaporation, B.t.u.

24. Use of Steam Tables.—Tables, giving values of the properties —enthalpy, internal energy,[1] entropy, volume, pressure, and temperature of steam—are provided to eliminate the long tedious calculations

[1] Steam tables by Keenan do not include internal energy; tables by Keenan and Keyes include internal energy.

necessary to obtain these properties. All tables are based on 1 lb. of steam or water, and the starting point, as explained before, is 1 lb. of water at 32°F. The pressures listed are absolute pressures and the temperatures are degrees Fahrenheit.

The make-up of the steam tables by different authors is along the same general lines, so that with a good understanding of one standard set of tables, an engineer is able to make use of any other. For this explanation, the tables by Prof. J. H. Keenan are used, and the discussion is limited to the following:

1. Tables 1 and 2, for saturated steam.
2. Table 3, for superheated steam.
3. Mollier diagram, for saturated and superheated steam.

The following equations show the method of obtaining the values of the properties of water and steam, designating the table used in each case.

1. *Water.*
   Enthalpy $= h_f$ (Table 1 or 2).
   Internal energy $= h_f - APv_f$ (Table 1 or 2).
   Entropy $= s_f$ (Table 1 or 2).
   Volume $= v_f$ (Table 1 or 2).

As the term APv_f is very small, the internal energy may be used as h_f.

2. *Wet Saturated Steam.*
   Enthalpy $= h_f + xh_{fg}$ (Table 1 or 2).
   Internal energy $= (h_f + xh_{fg}) - APxv_g$ (Table 1 or 2).
   Entropy $= s_f + xs_{fg}$ (Table 1 or 2).
   Volume $= xv_g + (1 - x)v_f$ (Table 1 or 2).

As v_f and $(1 - x)$ are comparatively small, the term $(1 - x)v_f$ may be neglected; therefore, xv_g may be used for the volume of wet saturated steam.

3. *Dry Saturated Steam.*
   Enthalpy $= h_g$ (Table 1 or 2).
   Internal energy $= h_g - APv_g$ (Table 1 or 2).
   Entropy $= s_g$ (Table 1 or 2).
   Volume $= v_g$ (Table 1 or 2).
4. *Superheated Steam.*
   Enthalpy $= h$ (Table 3).
   Internal energy $= h - APv$ (Table 3).
   Entropy $= s$ (Table 3).
   Volume $= v$ (Table 3).

Example 2-3.—Assuming a constant pressure of 155.3 lb. per square inch, ga., and barometer 29.92 in. of mercury, determine the enthalpy, internal energy, entropy, and volume for 1 lb. of (*a*) water at saturation temperature, (*b*) 98 per cent dry steam, (*c*) steam at 500°F.

Solution.

$155.3 + 0.491 \times 29.92 = 170$ lb. absolute.
Saturation temperature at 170 lb. = 368.42°F.

a. Water.
$h_f = 341.03$ B.t.u. (Table 2, for both the enthalpy and internal energy).
$s_f = 0.5268$ (Table 2).
$v_f = 0.01821$ cu. ft. per pound (Table 2).
b. Steam.

$$x = 98 \text{ per cent.}$$
$$h_f + xh_{fg} = 341.03 + 0.98 + 854.4 = 1178.3 \text{ B.t.u.}$$
$$h_f + xh_{fg} - APxv_g = 1{,}178.3 - \tfrac{1}{778} \times 170 \times 144 \times 2.618 = 1{,}096.3$$
$$s_f + xs_{fg} = 0.5268 + 0.98 \times 1.0318 = 1.5378$$
$$xv_g = 0.98 \times 2.671 = 2.618 \text{ cu. ft.}$$

All values from Table 2.
c. Superheated Steam.

$$t_s = 500°\text{F.}$$
$$h = 1{,}271.0 \text{ B.t.u.}$$
$$h - APv = 1{,}271.0 - \tfrac{1}{778} \times 170 \times 144 \times 3.229 = 1{,}169.2 \text{ B.t.u.}$$
$$s = 1.6435$$
$$v = 3.229 \text{ cu. ft.}$$

All values taken from Table 3.

Example 2-4.—Water at 125°F. enters a boiler which generates steam at 225 lb. per square inch, absolute pressure, and a quality of 98.7 per cent. The wet steam passes through a superheater and is superheated 175° at constant pressure. Find the heat added per pound, and the change in volume per pound (*a*) in the boiler, and (*b*) in the superheater.

Solution.—*a. Change in Enthalpy = Heat Added.*
At 225 lb. and 98.7 per cent quality

$$h_f + xh_{fg} = 366.1 + 0.987 \times 833.2 = 1{,}188.1 \text{ B.t.u.}$$
At 125°F. (Table 1) $h_f = 92.87$

Heat added per pound in boiler 1,095.23 B.t.u.
$t_s = t + 175 = 391.81 + 175 = 566.81°\text{F.}$
At 225 lb. and 566.81°F. $h = 1{,}301.36$ B.t.u.
$h_f + xh_{fg} = 1{,}188.1$

Heat added per pound in superheater = 113.26 B.t.u.

b. Change in Volume.
At 225 lb. and 98.7 per cent quality

$$xv_g = 0.987 \times 2.0393 = 2.01 \text{ cu. ft. per pound.}$$
At 125°F. (Table 1) $v_f = 0.01622$

Increase in volume in boiler = 1.99378 cu. ft. per pound.
At 225 lb. and 566.81°F. $v = 2.607$
$xv_g = 2.01$

Increase in volume in superheater = 0.597 cu. ft. per pound.

25. The Mollier Diagram.—The Mollier diagram supplements Tables 1, 2, and 3 and is especially useful in the solution of problems involving adiabatic or constant enthalpy changes. In the Mollier diagram the vertical lines represent units of entropy and the horizontal lines represent enthalpy per pound of steam. The absolute pressure lines extend upward to the right. The saturation line divides the diagram into two areas, the region of superheated steam and the region of wet saturated steam. In the superheat region there are parallel

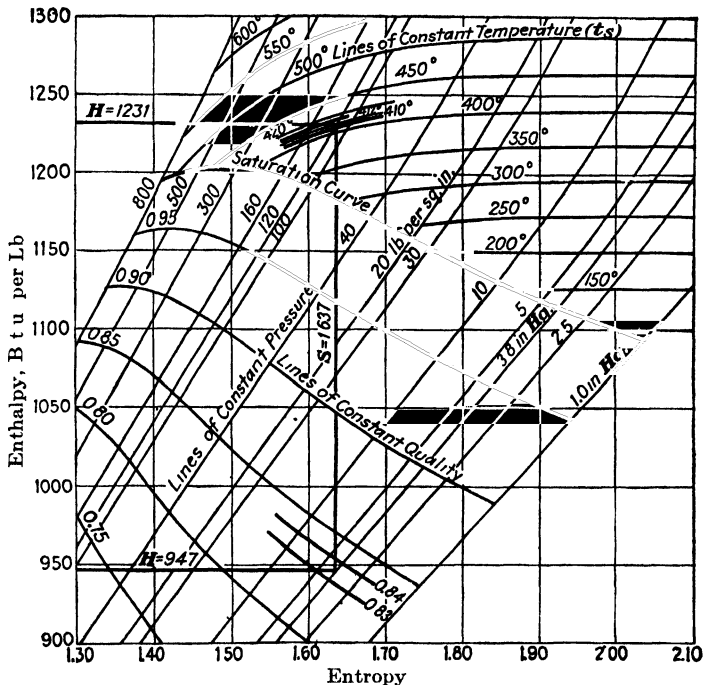



FIG. 6.—Mollier diagram.

lines representing steam temperatures and another set of parallel lines for degrees of superheat, while in the saturated region the parallel lines represent constant moisture, in percentage.

Any point on the diagram represents steam at a definite condition of pressure, temperature or quality, entropy, and enthalpy. If the point is located, these properties can be determined from the diagram. Moreover, if two of the properties are known, the point can be located on the diagram and the other properties determined.

Example 2-5.—Using Mollier diagram, find the properties of steam at 120 lb. per square inch, absolute pressure, and a temperature of 412°F. (An illustration of the solution is shown by Fig. 6, a skeleton Mollier diagram.)

Solution.—At the intersection of the lines, $p = 120$ lb. and $t_s = 412°$F., is located a point which indicates the state of the steam. The horizontal line through this point indicates the enthalpy to be 1,231 B.t.u. per pound, and the vertical line through it gives the entropy as 1.637.

Problems involving adiabatic changes are readily solved, using the Mollier diagram. For the conditions given, the initial state is located, which thus determines the entropy. Since an adiabatic change occurs at constant entropy, the intersection of the entropy line with the final pressure line locates, on the diagram, the point representing the final state. Likewise, with constant enthalpy, the horizontal line representing enthalpy is used instead of the entropy line.

The following example is solved using (1) the Mollier diagram, and (2) the steam tables 1, 2, and 3.

Example 2-6.—Steam (of Example 2-5) expands adiabatically to a vacuum of 26 in. of mercury; barometer 29.8 in. of mercury. Determine the properties of the steam at the final condition.

Solution.—*a.* By Mollier diagram (see Fig. 6).
Final pressure

$$p_2 = 29.8 - 26 = 3.8 \text{ in. of mercury, abs.}$$

From Example 2-5

$$h_1 = 1{,}231 \text{ B.t.u. per pound}$$
$$s_1 = 1.637$$

The intersection of $s = 1.637$ and $p = 3.8$ in. of mercury or 1.86 lb.; gives the point indicating the final condition of the steam.
At this point are found the following:

$$x_2 = 0.835 \text{ or } 83.5 \text{ per cent}$$
$$h_2 = 947 \text{ B.t.u. per pound}$$

Solution.—*b.* Using tables.
For state 1, using Table 3:

At $p_1 = 120$ lb., $t_s = 412$, is found $h_1 = 1{,}230.28$ B.t.u. per pound
and $s_1 = 1.6358$

As the expansion is adiabatic,

$$s_1 = s_2$$

and as the final condition is wet steam

$$s_2 = s_f + xs_{fg}$$

From Table 1, for

$$p_2 = 3.8 \text{ in. of mercury}$$
$$s_2 = 0.1704 + x_2(1.7549)$$

Therefore,

$$1.6356 = 0.1704 + 1.7549x_2$$

and

$$x_2 = 0.835 \text{ or } 83.5 \text{ per cent}$$
$$h_2 = h_f + x_2h_{fg} = 91.28 + 0.835 \times 1{,}023.14 = 945.6 \text{ B.t.u. per pound}$$

26. Steam Calorimeters.—To determine the enthalpy of wet saturated steam, it is necessary to know the quality. The apparatus used to determine the quality of steam is called a steam calorimeter. There are several different types of steam calorimeters.

27. Barrel Calorimeter.—The barrel calorimeter is the simplest and least accurate device of this kind used. It consists, simply, of a barrel or tank containing a known weight of water at a known temperature. The steam, of which the quality is desired, is discharged into the water and condensed, care being taken to stop the steam flow before the temperature rises to or above 180°F. The final weight and temperature are taken, and the quality determined as follows:

The heat given up by the steam equals the heat absorbed by the water.

FIG. 7.—Separating calorimeter.

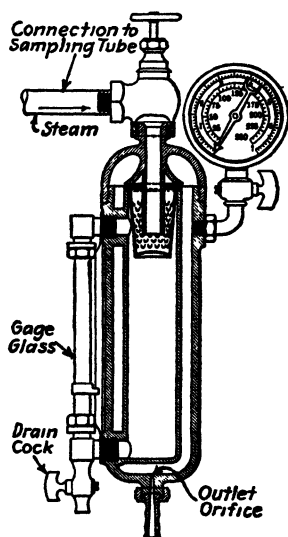
$$W_s(h_{f1} + x_1h_{fg_1} - h_{f2}) = W_w(h_{f2} - h_{f3})$$

hence,

$$x_1 = \frac{W_w(h_{f2} - h_{f3}) - W_s(h_{f1} - h_{f2})}{W_sh_{fg_1}} \qquad (45)$$

in which

p_1 = steam pressure, lb. per square inch absolute.
t_2 = final temperature of water, °F.
t_3 = initial temperature of water, °F.
W_s = weight of steam condensed, lb.
W_w = initial weight of water, lb.

28. Separating Calorimeter.—In the separating calorimeter (Fig. 7) the steam is discharged into a reversing cup where the water is thrown out mechanically and collected in an inner chamber. A gage glass with a scale calibrated in hundredths of a pound indicates the weight of

water collected. The dry steam passes through the jacket and out through an orifice of known diameter, and the total amount can be calculated or condensed and weighed. Ordinarily, there is a steam gage provided, to indicate the pressure before the orifice, and it is usually calibrated to indicate the pounds of dry steam flowing through the orifice in 10 min. Otherwise Napier's formula, as follows, is used to calculate the flow of dry steam:

$$W_s = \frac{pa}{70}$$

W_s = weight of dry steam per second, lb.
p = pressure, lb. per square inch absolute.
a = area of orifice, sq. in.

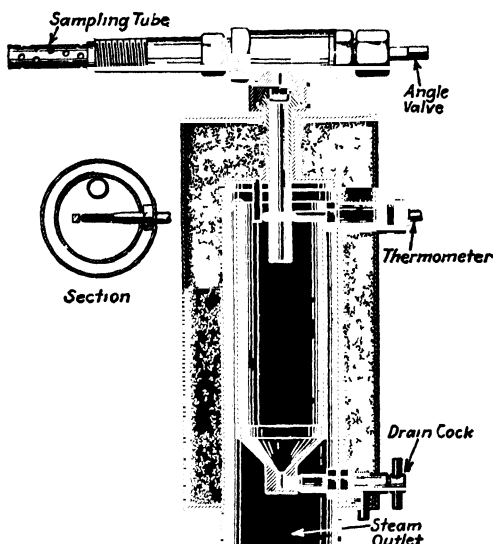



FIG. 8—Ellison throttling calorimeter.

It is more accurate to condense the dry steam and obtain its weight. The quality is then determined by the following equation:

$$x = \frac{W_s}{W_w + W_s} \times 100$$

where

W_s = weight of dry steam per second, lb.
W_w = weight of moisture collected per second, lb.
x = quality, per cent.

29. Throttling Calorimeter.—The throttling calorimeter is the most accurate of the commonly used calorimeters when used for steam

containing not more than 4 per cent moisture. There are many different designs of throttling calorimeters, but they are all similar in principle. In the calorimeter shown in Fig. 8, the wet steam enters the perforated sampling pipe and flows horizontally to the orifice, where it expands to atmospheric pressure without loss of heat. The steam makes an S turn in the inner chamber and flows downward through the jacket, escaping to the atmosphere. The connecting pipe and calorimeter chamber are well insulated to reduce radiation losses. A steam-pressure gage gives the pressure of the wet steam, and a thermometer gives the temperature of the steam inside the calorimeter. This throttling expansion is at constant enthalpy, which results in the formation of superheated steam below the orifice. The enthalpy of the wet steam is equated to the enthalpy of the superheated steam and the quality is obtained; thus,

$$h_s = h_f + x_1 h_{fg}$$

solving,

$$x_1 = \frac{h_s - h_f}{h_{fg}} \tag{46}$$

in which

h_s = enthalpy of steam (Table 3, steam tables) in calorimeter chamber, B.t.u. per pound.
h_f = enthalpy of liquid of steam, in line, B.t.u. per pound.
h_{fg} = enthalpy of vaporization of steam, in line, B.t.u. per pound.
x_1 = quality of steam in line.

Example 2-7.—Wet saturated steam at a pressure of 130 lb. per square inch, abs. flows into a throttling calorimeter where the resulting temperature is 260°F. Determine the initial quality.

Solution.—*a*. Mollier diagram.

At the intersection of p = 14.7 lb. absolute and t_s = 260°F.,

$$h_s = 1{,}173 \text{ B.t.u.}$$

At the intersection of h = 1,173 and p = 130 lb. abs.,

$$x_1 = 0.98.$$

Solution.—*b*. From tables.

From Table 3, p = 14.7 lb., t = 260.

$$h_s = 1{,}173.4 \text{ B.t.u.}$$

From Table 2, p = 130 lb.

$$h = 318.73 + 872.4x_1$$

Since

$$h = h_s$$

$$318.73 + 872.4x_1 = 1{,}173.4$$

$$x_1 = \frac{1{,}173.4 - 318.73}{872.4}$$

$$= 0.98 \text{ or } 98 \text{ per cent}$$

Problems

1. Convert the following temperatures from the Centigrade to the Fahrenheit scale: (*a*) 0°, (*b*) −50°, (*c*) 345°, and (*d*) 100°C.

2. One pound of petroleum has a heating value of 18,500 B.t.u. How many foot-pounds of work per pound of fuel are produced if the power plant burning this oil has a plant efficiency of 21.5 per cent?

3. An engine develops 340,000 ft.-lb. of work per minute. If its efficiency is 12.5 per cent, how much heat is supplied, B.t.u. per hour, and how many B.t.u. per hour are rejected?

4. A Diesel engine develops 3,960,000 ft.-lb. for each 10 lb. of fuel oil burned. The heating value of the oil is 18,350 B.t.u. per pound. What is the thermal efficiency of the engine?

5. Air at a pressure of 14.5 lb. per square inch absolute and a temperature of 85°F. is compressed from a volume of 10 cu. ft. to a final pressure of 85 lb. per square inch absolute and temperature of 197°F. What is the final volume?

6. Using two methods, find the value of R for the following gases: hydrogen, nitrogen, air, helium and oxygen.

7. Using two methods, find the values of R for carbon monoxide, carbon dioxide, sulphur dioxide, methane and acetylene.

8. Find the volume of 15 lb. of carbon dioxide at a pressure of 25 lb. per square inch absolute and a temperature of 56°C.

9. A tank contains 6,000,000 cu. ft. of helium at 14.7 lb. per square inch absolute and at a temperature of 100°F. How large a tank would be required to hold the same weight of helium at 2,500 lb. per square inch and at the same temperature?

10. An automobile tire may be assumed to have a mean diameter of 28 in., and an internal width of 5.25 in. It contains air at 32 lb. per square inch gage, and at 70°F.; barometer 30 in. of mercury. On the road the tire is heated to 120°F. What is the resulting pressure, assuming the volume to remain constant?

11. A closed tank is filled with 100 cu. ft. of carbon dioxide at 25 lb. per square inch absolute. If 200 B.t.u. are added to the enclosed gas, what is the resulting temperature and pressure? (w = 20 lb.)

12. Five cu. ft. of air are compressed isothermally from a pressure of 45 lb. absolute and a temperature of 70°F. to a pressure of 180 lb. absolute. What is: (*a*) the final volume; (*b*) final temperature; (*c*) work of compression; (*d*) increase in internal energy; (*e*) B.t.u. to be abstracted from the cylinder?

13. Solve Problem 12, considering the compression to be adiabatic.

14. One pound of air at 200 lb. per square inch absolute and a volume of 2 cu. ft. expands until its pressure is 20 lb. per square inch absolute and its volume is 16 cu. ft. Find (*a*) the value of n; (*b*) work done during expansion; (*c*) change in internal energy; (*d*) heat added from cylinder walls during expansion.

15. Two cubic feet of air at 85°F. and 13.9 lb. absolute pressure are compressed adiabatically to 115 lb. absolute. Find the final temperature, final volume, work in foot-pounds, change in internal energy, and the heat flow.

16. Five pounds of carbon dioxide at 14.7 lb. absolute and 75°F. are compressed to 85 lb. absolute according to the law $PV^{1.21} = C$. At the end of compression, the temperature is increased 75° more at constant pressure. Find the temperature at the end of compression, and the total heat change.

17. If $3\frac{1}{2}$ lb. of air having a volume of 5 cu. ft. are heated at a constant pressure of 150 lb. absolute until the volume is doubled, how much work is done?

18. If $3\frac{1}{2}$ lb. of air, at a pressure of 150 lb. absolute and a volume of 10 cu. ft., are expanded isothermally to a pressure of 50 lb. absolute, find the final volume and temperature.

19. Find the change in internal energy when carbon dioxide expands from a pressure of 220 lb. absolute to a pressure of 18 lb. absolute. Initial and final volumes are 3 and 55 cu. ft., respectively.

20. Find (*a*) the enthalpy of the liquid, (*b*) enthalpy of vaporization and (*c*) enthalpy of 1 lb. of dry saturated steam at 100 lb. per square inch, gage pressure, when the barometer reading is 30 in. of mercury.

21. Determine the increase in volume of 1 lb. of water at 100°F. on being changed into (*a*) dry saturated steam at 225 lb. per square inch absolute pressure, and (*b*) steam, 96 per cent dry, at same pressure.

22. Determine the change in entropy in part (*a*) of Problem 21.

23. One pound of steam, 98.5 per cent dry and at a pressure of 155 lb. per square inch absolute, expands adiabatically to a pressure of 50 lb. per square inch absolute. Determine the quality after expansion.

24. If heat is added to water, initially at 80°F. and 161 lb. per square inch absolute pressure, until it becomes superheated steam at 200°F. above saturation temperature, (*a*) how many B.t.u. are required, and (*b*) what is the increase in volume?

25. Wet steam of unknown quality undergoes a constant heat expansion during a drop in pressure from 200 lb. per square inch absolute to 20 lb. per square inch absolute and 250°F. Find the enthalpy per pound and the quality of the wet steam.

26. Steam expands in a throttling calorimeter from a line pressure of 210 lb. per square inch gage to 5 lb. per square inch gage. Find the initial quality of the steam if the temperature in the calorimeter is 285°F. and the barometer reads 29.4 in. of mercury.

27. Solve Problem 26 when the initial pressure is 305 lb. per square inch gage and the calorimeter temperature is 315°F.

28. In a separating calorimeter the pressure is 175 lb. per square inch absolute. The moisture collected in 15 min. is 0.22 lb. Find the quality of the steam if the diameter of the dry steam orifice is 0.08 in.

29. Solve Problem 28 if the calorimeter pressure is 124 lb. per square inch absolute and the moisture collected in 10 min. is 0.18 lb.

30. One pound of air under 150 lb. per square inch absolute pressure and 200°F. undergoes a constant pressure process after which its volume is 10 cu. ft. Find (*a*) change in internal energy, (*b*) heat flow.

31. A gas mixture has the following composition by weight: CO 24 per cent, CO_2 6 per cent, N_2 59 per cent, and SO_2 11 per cent. Find (*a*) value of R_m, and (*b*) value of T_m if 0.10 lb. of the mixture is confined to a volume of 1.5 cu. ft. at a pressure of 25 lb. per square inch absolute.

32. Using the gas analysis of Problem 31, find (*a*) value of Cpm, and (*b*) value of Q if 10 lb. of the mixture are heated at constant pressure from 50 to 360°F.

33. Steam at 90 lb. per square inch absolute, 99 per cent dry, expands adiabatically to 7 lb. per square inch absolute. Determine work done in foot-pounds.

34. Steam at 328 lb. per square inch absolute, volume of 1.81, is cooled at constant volume to 225 lb. per square inch absolute. Determine heat abstracted.

35. Steam at 190 lb. per square inch absolute, 95 per cent dry, is throttled (constant h) to 4 lb. per square inch absolute. Determine change in entropy

CHAPTER III

FUELS AND COMBUSTION

30. General.—Fuels are either gaseous, liquid, or solid organic compounds which contain elements that are capable of uniting with the oxygen of the air with sufficient rapidity to produce heat, at a comparatively high temperature. This process is known as *combustion.* There are certain fuels, such as graphite, which can unite with oxygen, but at such a slow rate that, for burning in the ordinary boiler furnace, they cannot be considered as fuels for the production of power. An ideal fuel would be a gas composed entirely of combustible elements, as such a fuel can be easily mixed with air in proper proportions, readily brought to the ignition temperature and burned without difficulty. In order for combustion to take place, the ignition temperature (Table 3-7, page 65) of the combustible elements of the fuel must be attained in the presence of oxygen. The combustible elements of the common fuels are carbon, hydrogen, and sulphur, of which the carbon and hydrogen are the most active. In solid fuels, certain constituent materials, such as ash, nitrogen and moisture, possess no heat-producing qualities, are inert and even hinder combustion.

31. Classification of Fuels.—Fuels may be classified according to their physical character as:

1. Solid.
2. Liquid.
3. Gaseous.

Of all the fuels, coal, in its various classes, is the most important one used for power purposes. Other solid fuels, such as wood, peat, charcoal, coke, straw, corn, bagasse, and tanbark, are used in certain localities.

Liquid fuels, consisting chiefly of petroleum, the residue after the lighter oils have been removed, follow coal in importance, as a source of heat for steam generation. Coal tar, a by-product of the coking process, is used to a limited extent. Colloidal fuel, an emulsion of powdered coal and oil, is used in a few cases by the U. S. Navy. Table 3-4 (page 57) gives the chemical composition of various fuel oils. Petroleum and its distillates furnish the fuel for almost all of the

internal-combustion engine power in this country, which amounts to about 50 per cent of the total mechanical power generated.

Gaseous fuels are either natural or artificial. Natural gas, from wells, is commonly used for domestic purposes, and, when available, is very desirable as a power fuel. The artificial gases, such as coke-oven, blast-furnace or producer gas, are used as fuels for steam boilers or internal-combustion engines mainly by industries which have these gases as by-products. Heat values and various chemical constituents of the commonly used gaseous fuels are given in Table 3-5 (page 58).

32. Solid Fuels Other than Coal.—Wood fuel, as used commercially for steam generation, consists of sawdust, "hogged wood," slabs, shavings, etc., which are usually a waste product from some industrial process. Hogged wood is mill refuse that has been shredded or cut to small pieces of more or less uniform size by a machine called a "hog," which facilitates efficient handling and burning. Freshly cut wood contains from 20 to as high as 60 per cent of moisture, but when air dried the moisture content may be reduced to as low as 20 per cent.

Wood is ordinarily classified as (1) hard wood, such as oak, maple and walnut, and (2) soft wood, such as pine, fir, hemlock, spruce and elm. The heat value and the ease of combustion of wood depends, to a large extent, on the presence of resins, gums and "tarry" oils. Because of this fact the heat value per pound of soft wood is higher than that of hard wood. The heat values of various wood fuels, as listed in Table 3-6 (page 58), are given on the dry basis, which is common practice. It should be noted that wood fuel, when burned with a high moisture content, has a smaller portion of its potential heat available for steam generation. This is due to the fact that all of the moisture must be evaporated and superheated to the temperature of the escaping gases. Such absorption of heat reduces the gas temperature and, thereby, reduces the rate of heat flow to the water and steam inside the boiler.

Bagasse is the refuse from the sugar mills, that is, the cane fiber remaining after the juices have been extracted. It has a low ash content and, when dry, is a very good fuel.

Tanbark or "spent tan" is the fibrous remains from the ground oak or hemlock bark employed as a leather-tanning agent. When used as a steam-boiler fuel it is usually supplemented with a small amount of coal, to keep the volatile gases ignited. Table 3-6 (page 58) gives the usual combustion data for various fibrous fuels.

33. Coal.—The origin of coal gives rise to many perplexing theories. It is generally believed, however, that in prehistoric times vegetable

matter was gradually accumulated in thick and extensive beds. And in the course of geological change these beds became subjected to enormous pressure and excessive temperature over long periods of time, the effect of which was a destructive distillation, giving coal as we now find it.

TABLE 3-1.—COAL CLASSIFICATION BY RANK

Rank	Subgroup	Limits of fixed carbon or B.t.u. on mineral-matter free basis
I. Anthracite.	1. Meta anthracite	Dry, fixed carbon 98 per cent or more
	2. Normal anthracite	Dry, fixed carbon 92 up to 98 per cent
	3. Semianthracite	Dry, fixed carbon 86 up to 92 per cent *
II. Bituminous.	1. Low volatile	Dry, fixed carbon 77 up to 86 per cent
	2. Medium volatile	Dry, fixed carbon 69 up to 77 per cent
All coals below this group, dry, fixed carbon less than 69 per cent		
	3. High volatile *A*	Moist, B.t.u. 14,000 or more
	4. High volatile *B*	Moist, B.t.u. 13,000 up to 14,000
	5. High volatile *C*	Moist, B.t.u. 11,000 up to 13,000†
III. Subbituminous....	1. *A*	Moist, B.t.u. 11,000 up to 13,000‡
	2. *B*	Moist, B.t.u. 9,500 up to 11,000
	3. *C*	Moist, B.t.u. 8,300 up to 9,500
IV. Lignitic..........	1. Lignite	Moist, B.t.u. less than 8,300§
	2. Brown coal	Moist, B.t.u. less than 8,300‖

* Non-agglutinating.
† Either non-agglutinating or non-weathering.
‡ Weathering and non-agglutinating.
§ Consolidated.
‖ Unconsolidated.

Coal is placed in different classes or ranks, according to the chemical and physical properties resulting from its geological formation. The U. S. Bureau of Mines uses the term "rank" to signify the stage in this process between peat, the premature coal, and graphite or that coal which is completely carbonized. The names which apply to the various ranks, in order, and starting with the lowest, are:

1. Peat.
2. Lignite.
3. Subbituminous.
4. Bituminous.

5. Semibituminous.
6. Semianthracite.
7. Anthracite.
8. Graphite.

It is difficult to place exactly all coals in the proper rank. The classification by Selvig, Ode, and Fieldner, 1934 *A.I.M.E. Proceedings* (Table 3-1), gives an exact method, using only four ranks with subgrouping in each.

The term agglutinate is used to designate the cementing process that occurs during the combustion of a caking coal. The Bureau of Mines* obtains agglutinating indices of coal samples by mixing fine sand with pulverized coal in definite proportions. The mixture is heated severely without access to air, and the volatile gas is driven off. The strength of the resulting button of coke is a measure of the agglutinating property of the coal. The weathering test consists of air drying a sample of about 1-in. lumps for 24 hr. The coal is then immersed for 1 hr., then air-dried for another 24 hr. and the amount of disintegration obtained by separation through a sieve.

The preceding table is on a mineral-matter free basis. This is a little different from the ash-free basis. Parr determines the mineral matter of coal as follows:

$$\text{Mineral matter} = 1.08A + \frac{22}{40}S$$

An approximate simpler equation, generally satisfactory, for the determination of mineral matter of coal, follows:

$$\text{Mineral matter} = 1.1A$$

In common usage, however, coal is classified by the eight ranks from peat to graphite. Hence a brief discussion of the characteristics of each rank follows.

Peat and *graphite* are not used as fuels for power generation; peat, because of its low heat value, high moisture content and difficulty of handling and storing; graphite, because of the difficulty of ignition and burning. The greater quantity of power fuels fall within the various grades of bituminous coals.

Lignite is a coal of low rank and of more recent geological origin than the better coals, such as bituminous, etc., as is indicated by the above classification. Because of its high moisture content (30 to 50 per cent) and low heat value, it is not especially suited for use in efficient generation of steam. Lignite is brown in color, and, when

* Reprinted from U. S. Bur. Mines *Report of Investigations* 3296.

exposed to the air, it disintegrates into small flakes having the texture of crumbled clay. Because of this characteristic it can be stored for only a short time and is shipped with some difficulty. Lignite is therefore, commonly burned only in the vicinities where it is mined. It is burned on hand-fired grates, mechanical stokers, and in pulverized form.

Bituminous coals, including subbituminous, bituminous and semibituminous, are by far the most generally used fuels for power generation. Subbituminous coal is somewhat like lignite in its tendency to slack, but its color is black, it lacks the "woody texture" of lignite and is slightly more desirable as a power fuel. *Bituminous* covers a variety of coals which possess but few distinguishing characteristics, making a classification difficult. They are either "caking" or "non-caking"; the former having the property of forming a fused mass of coke when burned, while the latter burn freely, that is, without a fusing action. In this rank are found the gas coals, cannel coal and certain trade coals known as "block" and "splint." The non-caking coals are found chiefly in the middle western and western states, while the caking coals come from the Pittsburgh district.

The *semibituminous* coals approach anthracite in characteristics, but, for the most part, they possess extremely high caking properties. The better grades have a higher heat value than other coals, and they burn easily, without smoke. Therefore, they are highly desirable for steam-power purposes.

In the United States the available supply of *semianthracite* is comparatively small, northern Pennsylvania and central Virginia being the location of the few fields. Semianthracite burns freely, with a short flame, and has a tendency to swell without caking. The heat value is high and it does not form clinkers at ordinary furnace temperatures. This coal, however, is relatively unimportant, due to the limited supply and the resulting high cost.

Anthracite is an exceptionally good coal for domestic purposes because of its smokeless combustion, low ash content and cleanliness in handling, but for power purposes its cost is almost prohibitive. Due to recent improvements in combustion equipment the "culm" (refuse from anthracite screenings) and "bone coal" (low-grade anthracite) are now being burned in many power plants. Anthracite burns with an exceptionally short, yellow flame which changes to blue after its small amount of volatile matter has been burned. The outstanding physical properties of this coal are extreme hardness and a black lustrous surface. The best known deposits of anthracite are found in three small fields in eastern Pennsylvania, and the supply is

estimated at less than 5 per cent of the total reserve of unmined coal in the United States.

Figure 9 shows the locations of the various main deposits of coal in the United States.

The *size of coal,* when being burned, has a decided effect on the efficiency and capacity of a boiler. Consequently, it is very desirable to use coal of a size that is conducive of the best and most economical operation. Plants equipped with crushers probably have the best solution to the coal-size problem. Coal is sold, commercially, in various sizes ranging from fine waste to large lumps. Table 3-2 gives

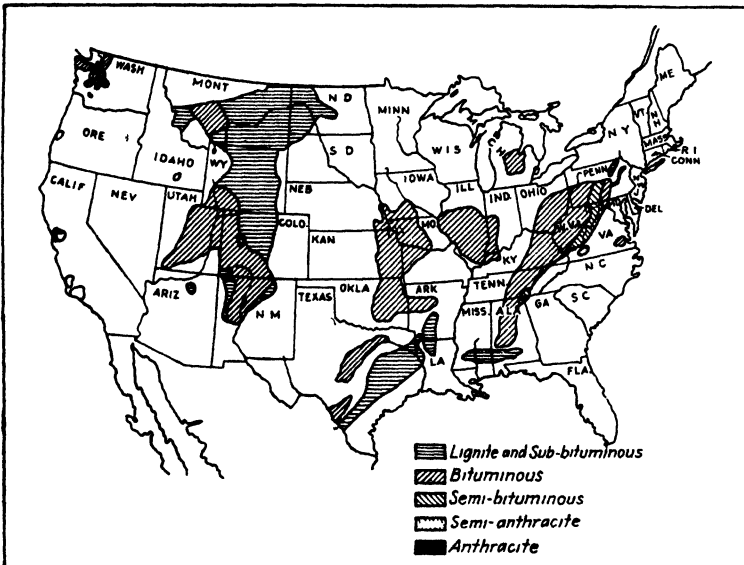



Fig. 9.—Map showing distribution of coal deposits in the United States.

the well-known standards generally used. In some parts of the country, however, certain local standards often prevail.

Coal is often *washed* for the purpose of removing certain impurities, such as earthy matter, slate, and sulphur. In general, the ash content of coal increases with the degree of fineness. Washing tends to reduce the amount of fine particles and thereby reduces the ash content.

To *purchase* coal economically, and to get the most suitable grade for a particular installation, requires the consideration of many factors. Boiler furnaces are built to burn a particular coal efficiently, and the range of deviation is comparatively small. Having determined the rank of coal to be used, it then remains to specify (often by written agreement) the limits of such characteristics as (1) ash content,

(2) moisture, (3) size, (4) heat value per pound and (5) the nature and amount of sulphur content. The latter is essential, to some extent, in determining the clinkering and caking characteristics of the coal. Also, in connection with pulverized coal burning, the *ash fusion temperature* is of especial significance.

TABLE 3-2.—COMMERCIAL COAL SIZES

Rank and trade name	Diam., in., which coal will pass	
	Through	Over
Anthracite (A.S.T.M. Std.)		
Broken	4½	3¼
Egg	3¼	2⅜
Stove	2⅝	1¼
Chestnut	1¼	¾
Pea	¾	½
Buckwheat No. 1	½	¼
Buckwheat No. 2	¼	3/16
Buckwheat No. 3	3/16	3/32, 1/16
Culm	3/32, 1/16	
Eastern bituminous (A.S.M.E. Std.)		
Run of mine	As mined	
Lump	....	1¼
Nut	1¼	¾
Slack	¾	
Western bituminous (A.S.M.E. Std.)		
Run of mine	As mined	
Lump, 6 in	....	6
Lump, 3 in	....	3
Lump, 1¼ in	....	1¼
Nut, 3 in	3	1¼
Nut, 1¼ in	1¼	¾
Nut, ¾ in	¾	⅝
Screenings	⅝	

34. Fuel Analysis.—A fuel analysis is essential, particularly in connection with tests, and, also, to enable the engineer to determine scientifically the probable results when the fuel is burned. Solid fuels are generally considered as being composed of combustible matter. moisture and ash. The *combustible matter,* however, is not all combustible as it includes nitrogen and oxygen, neither of which can be burned.

The analysis of coal may be given as (1) *proximate analysis* or (2) *ultimate analysis;* the former giving those characteristics that may

be determined mechanically (mainly, by the application of heat), and the latter giving the chief chemical constituents. Table 3-3 gives

TABLE 3-3.—ANALYSES OF COAL SAMPLES FROM MINES IN THE UNITED STATES
(Compiled from U. S. Bureau of Mines Bulletins)

State	County	Rank	Analysis of coal as received									Heat value, B.t.u. per pound as received
			Proximate				Ultimate					
			Moisture	Volatile matter	Fixed carbon	Ash	Sulphur	Hydrogen	Carbon	Nitrogen	Oxygen	
Ala.	Bibb	Bit	2.71	34.67	57.43	5.19	1.26	5.29	79.71	1.42	7.13	14,157
Calif.	Monterey	Bit.	8.02	48.46	36.01	7.51	4.09	5.90	65.21	1.24	16.05	12,130
Colo.	El Paso	Subbit.	26.00	33.49	35.36	5.15	0.30	6.27	49.02	0.57	38.69	8,231
Colo.	Las Animas	Bit.	4.29	36.51	52.03	7.17	0.58	5.38	73.93	1.39	11.55	13,086
Colo.	Routt	Bit.	12.82	37.15	44.44	5.59	0.43	5.77	63.26	1.49	23.46	11,034
Ill.	Christian	Bit.	11.93	38.21	39.50	10.36	4.00	5.73	61.37	1.02	17.52	11,086
Ill.	Franklin	Bit.	7.64	31.37	50.03	10.96	1.06	5.06	66.21	1.45	15.26	11,687
Ill.	Perry	Bit.	10.76	33.10	47.37	8.77	0.84	5.38	64.49	1.36	19.16	11,524
Ill.	Williamson	Bit.	9.44	32.99	48.95	8.62	0.93	5.17	67.06	1.51	16.71	11,858
Ind.	Greene	Bit.	13.51	36.26	42.93	7.30	1.08	5.82	64.51	1.45	19.84	11,543
Ind.	Knox	Bit.	10.14	37.11	43.71	9.04	3.00	5.65	64.88	1.18	16.25	11,722
Ind.	Sullivan	Bit.	12.61	36.02	44.49	6.88	1.51	5.87	65.65	1.48	18.61	11,794
Ind.	Vermilion	Bit.	14.68	34.21	43.86	7.25	1.10	5.89	63.48	1.38	20.90	11,327
Ind.	Vigo	Bit.	13.10	36.83	41.73	8.34	2.60	6.01	63.37	1.29	18.39	11,484
Kan.	Cherokee	Bit.	5.09	34.47	52.17	8.27	3.34	5.23	71.81	1.20	10.15	13,082
Ky.	Letcher	Bit.	3.00	36.62	57.25	3.13	0.65	5.39	79.81	1.53	9.49	14,200
Ky.	Webster	Bit.	5.35	34.92	50.39	9.34	1.05	5.12	70.39	1.55	12.55	12,501
Md.	Alleghany	Semibit.	1.87	19.81	64.46	13.86	3.60	4.16	73.72	1.18	3.48	13,043
Md.	Alleghany	Semibit.	2.91	28.86	58.63	17.60	3.78	4.31	66.48	1.06	6.77	12,179
Mont.	Valley	Lig.	45.86	22.46	24.58	7.10	0.29	7.22	33.11	0.61	51.67	5,465
N. M.	Socorro	Bit.	3.19	37.96	45.10	13.75	0.82	5.25	68.36	1.32	10.50	12,307
Ohio	Columbiana	Bit.	4.79	34.83	52.95	7.43	1.79	5.39	73.99	1.38	10.02	13,192
Ohio	Jefferson	Bit.	2.21	39.63	47.50	10.66	4.96	5.28	71.04	1.34	6.72	13,025
Okla.	Latimer	Bit.	3.89	38.14	51.41	6.56	1.44	5.38	74.27	1.89	10.46	13,356
Pa.	Butler	Bit.	5.78	37.92	45.59	10.71	4.28	5.73	68.61	1.20	9.47	12,451
Pa.	Cambria	Semibit.	3.32	18.25	72.03	6.40	1.77	4.73	80.66	1.42	5.02	14,171
Pa.	Somerset	Semibit.	2.40	17.25	71.36	8.99	2.54	4.28	78.50	1.20	4.49	13,738
Pa.	Schuylkill	Anth.	3.33	3.27	84.28	9.12	0.60	3.08	81.35	0.79	5.06	13,351
Tenn.	Morgan	Bit.	2.19	32.53	58.25	7.03	2.52	5.28	78.24	1.61	5.32	14,108
Tex.	Webb	(Cannel) Bit.	4.42	46.04	30.53	19.01	2.07	5.76	59.32	1.15	12.69	11,065
Va.	Tazewell	Semibit.	3.00	20.25	72.24	4.51	0.47	4.65	84.00	1.17	5.20	14,605
Va.	Montgomery	Semianth.	2.20	9.96	81.63	6.21	0.47	3.65	84.95	0.92	3.80	14,272
Wash.	Lewis	Subbit.	27.82	33.66	29.74	8.78	0.99	6.68	45.35	0.87	37.33	7,942
W. Va.	McDowell	Semibit.	2.27	18.91	74.85	3.97	0.41	4.56	84.90	1.16	5.00	14,756
W. Va.	Mercer	Semibit.	2.03	20.29	73.73	3.95	0.76	4.73	84.85	1.12	4.59	14,814
Wyo.	Lincoln	Subbit.	20.98	35.09	40.75	3.18	0.59	6.22	57.09	0.90	32.02	9,896

the analysis of a large number of coals from various parts of the United States, as reported by the U. S. Bureau of Mines.

In *preparation* for a coal analysis, a representative gross sample is taken, usually from the car, pile or stoker hopper. It should be collected in small quantities from different parts of the pile, or at intervals if taken from the stoker hopper, and should be approximately 500 lb. total weight. The sample is then placed in a pile on a smooth concrete or iron floor and thoroughly mixed and crushed, after which it is piled in the shape of a cone. The cone is next flattened and divided into quarters and two diagonal quarters disposed of, thus reducing the quantity. The mixing, crushing and dividing are repeated until a small, finely crushed laboratory sample remains. This final quantity is then put into a glass pint or quart jar and sealed until the analysis is made.

35. Proximate Analysis.—This analysis consists of determining the percentages, by weight, of *moisture, volatile matter, fixed carbon* and *ash* in the coal. Frequently the sulphur content is desired and, if determined, is often reported with the proximate analysis. These items give the general information regarding the burning characteristics of the coal, and their sum, excluding the sulphur, should be 100 per cent. It should be noted that "proximate" does not refer to the degree of accuracy in making the test, but that it is simply a distinctive name.

The *moisture* determination consists of heating a 1-g. sample of finely crushed (60 mesh) coal in an oven through which air, dried by H_2SO_4, is circulated. The temperature is maintained at approximately 220°F. for 1 hr., after which the sample is accurately weighed. The loss in weight due to the heating process is taken as the moisture content. Some volatile matter is probably released in this operation, but the amount, if any, is small, and the comparative value of the results is unaffected. Moisture increases the boiler losses, also the shipping weight, and its presence is, therefore, undesirable. Often, however, coal of poor quality is more easily handled by a stoker or on grate if tempered, slightly, with moisture.

The *volatile matter* is determined by placing a porcelain capsule, containing 1 g. of the fine-coal sample, in a suitable laboratory furnace and allowing it to remain there for 7 min., at a temperature of approximately 1750°F. The capsule is closed with a cover, and after removal from the furnace it is allowed to cool before being weighed. The loss in weight minus the moisture is taken as the volatile matter. This constitutes a large portion of the fuel (see Table 3-3) and has a decided effect on its burning characteristics. A large furnace is required to burn coals high in volatile matter, if smokeless and efficient combustion is to be attained.

The *fixed carbon* is considered as the combustible remaining in the coal after the distillation of all the volatile matter. The percentage of fixed carbon is found by difference, following the ash determination, and is equal to 100 − (moisture + volatile matter + ash). Coals high in fixed carbon, such as anthracites, require comparatively small furnaces for successful burning.

In determining the *ash or refuse* content, the sample from the moisture determination is heated to redness and completely burned. The residue is ash. Ash not only reduces the combustible per pound of coal, but complicates the burning process.

36. Ultimate Analysis.—The ultimate analysis gives the percentages by weight of the carbon, hydrogen, sulphur, oxygen, nitrogen and the ash in the coal, and their sum should equal 100 per cent. The process of making an analysis of this kind is a matter of chemistry and should require the services of an experienced chemist. An ultimate analysis gives not only data to be used in boiler test calculations and in the calculation of the approximate heat value of the fuel, but it gives general information that has especial significance to the experienced engineer. Analyses, however, do not give all the information necessary for determining the true fuel value of a coal. The best test of any fuel is to burn it under actual operating conditions.

Either analysis is generally reported on the basis of coal

1. "As received" or "as-fired," or
2. "Moisture free" or "dry," or
3. "Moisture and ash free" or "combustible."

The former terms, in all cases, are used by the Bureau of Mines, and the latter are those given by the A.S.M.E. Power Test Code, which are the ones most generally used in practice.

For test purposes, engineers usually prefer the analyses to be given on the as-fired basis, as it then shows the condition in which the coal was fed to the furnace. Regardless of the basis used, the analyses may be converted from one to the other by a simple calculation. Thus, the various items on an as-fired basis may be changed to the dry basis by dividing each value by 1 − (proportional weight of moisture). Likewise, it can be changed to a combustible basis by dividing by 1 − (combined proportional weights of ash and moisture). The following example shows the conversion from one basis to another.

Example 3-1.—The ultimate analysis of a sample of Sullivan County, Ind., coal is given on the as-fired basis, in column *a* of the following table. It is required to convert these values to (1) the as-fired basis, giving the moisture as a separate quantity, (2) the dry basis, and (3) the combustible basis.

Solution.—The results are calculated by the method discussed and tabulated in the following:

Constituents	Coal as fired		Coal, dry	Coal, combustible
	a	*b*	*c*	*d*
Carbon	65.65	65.65	75.12	81.53
Hydrogen	5.87	4.47	5.12	5.55
Sulphur	1 51	1.51	1.73	1.88
Oxygen	18.61	7.40	8.47	9.20
Nitrogen	1.48	1.48	1.69	1.84
Ash	6.88	6.88	7.87	
Moisture (free)	......	12.61*		
Total	100.00	100.00	100.00	100.00

* From the proximate analysis.

It should be noted that, in the above example, the value for hydrogen and oxygen, in column *a*, includes the hydrogen and oxygen of the free moisture, as given by the proximate analysis. Column *b* shows the values corrected for the free moisture which is given as a separate quantity. The correction is made by subtracting $\frac{1}{9}$ of the moisture ($\frac{1}{9} \times 12.61$) from the hydrogen (5.87) and $\frac{8}{9}$ of the moisture ($\frac{8}{9} \times 12.61$) from the oxygen (18.61). This is done by reason of the fact that moisture is composed of 8 parts of oxygen and 1 of hydrogen, making a total of 9 for the whole quantity.

37. Ash Analysis.—An ash analysis for the determination of the unburned combustible and the moisture is frequently necessary in connection with boiler tests. The method of collecting the sample is similar to that for coal. Care should be taken to quench quickly the ash as soon as it is removed from the ash hopper. The combustible may be determined, by calculation, from the total weight of dry coal, total ash as computed from the ultimate analysis, and the total actual weight of dry ash. The combustible in the grate siftings is found by the method used for coal. The *ash fusion temperature* is often desirable, particularly in connection with pulverized coal. It is found by making cones of small quantities of the finely crushed, pure ash residue and melting them down in a Melter's furnace equipped with a high-temperature pyrometer.

38. Analysis of Oil Fuels.—Fuel-oil analyses consist of the determination of the chemical composition and certain physical character-

istics, such as the specific gravity, flash point and viscosity. As with other fuels, the chemical analysis is beyond the range of this book. It is, however, often desirable and may be used in a calculation of the heat value.

The *specific gravity* is a comparative ratio of the weight of a given volume of the liquid to the weight of an equal volume of water, and it is usually determined with a hydrometer, at a temperature of 60°F. The hydrometer scale may give the specific gravity direct or in degrees Baumé, which are arbitrary numbers advancing upward from 10 (specific gravity 1.00), for liquids lighter than water. The American Petroleum Institute (A.P.I.) has adopted this scale for use with oils. Specific gravity = 140 ÷ (130 + degrees Baumé).

The *flash point* is the temperature at which the oil gives off an inflammable gas. It is determined by heating a small quantity of the oil in an open or partly closed vessel (preferably the latter) and applying a flame to the distilled vapors.

Viscosity is a measure of resistance to flow. A definite value representing the viscosity is determined, usually, by either a Saybolt

TABLE 3-4.—CHEMICAL COMPOSITION AND CALORIFIC VALUE OF VARIOUS FUEL OILS

State and Name	Per cent, weight				Specific gravity	Flash point, °F.	Moisture, per cent	B.t.u. per pound
	C	H	S	O				
California, Kern River	86.36	11.27	0.89	......	0.967	216	....	18,562
California, Kern River	85.04	11.52	1.94	0.10*	....	...	1.40	17,871
California, Kern River	81.52	11.51	0.55	6.42*	.....	230	....	18,667
Texas, Beaumont	84.60	10.90	1.63	2.87	0.924	180	....	19,060
Texas, Beaumont	83.30	12.40	0.50	3.80	0.926	216	....	19,481
Texas, Beaumont	87.15	12.33	0.52	......	0.908	370	....	19,338
Pennsylvania	84.90	13.70	....	1.40	0.886	...	....	19,210
West Virginia	84.30	14.10	....	1 60	0.841	...	....	21,240

* Includes N.

or an Engler viscosimeter, with the oil at 60°F. The principle of each is to measure the time required for the flow of a definite volume of oil through a special orifice. The Saybolt values are expressed as "Saybolt-seconds," which are time readings in seconds, while the Engler values are measured on a scale based on the ratio of the time required for the flow of 200 c.c. of water, at 60°F., through an orifice to the time required for the same quantity of oil, at the same temperature, to flow through the same orifice.

Table 3-4 gives characteristic analysis data for several fuel oils from different parts of the United States.

39. Analysis of Gas and Wood Fuels.—Gas fuels are chemically analyzed for certain constituent gas elements of which the characteristics are known. Table 3-5 gives average data from the analyses of a number of gases that are often used as power fuels.

TABLE 3-5.—COMPOSITION AND CALORIFIC VALUES OF VARIOUS GAS FUELS

Kind of gas	Per cent, volume							Heat value, B.t.u. per cu. ft.
	H	CH_4	C_2H_2	CO	CO_2	O_2	N	
Natural	1.7	94.16	0.30	0.55	0.29	0.20	2.80	1,000
Illuminating	39.78	45.16	6.38	7.04	1.08	0.06	0.50	730
Water (carburetted)	21.8	30.7	12.9	28.1	3.8	0.5	2.2	700
Coke-oven	53.2	34.8	2.0	6.0	2.0	....	2.0	620
Blast-furnace	3.0	.....	.....	27.5	10.10	....	59.4	100
Producer	2.80	5.56	.....	14.34	10.50	0.1	66.7	110
Oil	32.0	48.0	16.5	.....	.....	0.5	3.0	850

TABLE 3-6.—CHEMICAL COMPOSITION AND CALORIFIC VALUES OF VARIOUS FIBROUS FUELS[1]

Fuel	Per cent,[2] weight						B.t.u. per pound, dry basis
	C	H	O	N	Ash	Moisture	
Oak	50.16	6.02	43.36	0.09	0.37	.....	8,316
Ash	49.18	6.27	43.91	0.07	0.57	.....	8,480
Elm	48.99	6.20	44.25	0.06	0.50	.....	8,510
Beech	49.06	6.11	44.17	0.09	0.57	.....	8,591
Birch	48.88	6.06	44.67	0.10	0.29	.....	8,586
Fir	50.36	5.92	43.39	0.05	0.28	.....	9,063
Pine	50.31	6.20	43.08	0.04	0.37	.....	9,153
Bagasse, Cuba	43.15	6.00	47.95	....	2.90	51.50	7,985
Bagasse, Cuba	43.61	6.06	48.45	....	1.88	42.50	8,240
Bagasse, Porto Rico	44.21	6.31	47.72	0.41	1.35	43.50	8,386
Bagasse, Louisiana	.....	....	.....	....	....	54.00	8,370
Bagasse, Louisiana	46.03	6.56	45.55	0.18	1.68	.....	8,681
Tanbark	52.00	6.01	40.54	....	1.45	65.00	2,700, wet
Chips, chestnut wood	.....	....	.....	....	....	63.00	3,600, wet

Hemlock, spruce and cedar, not given in table, run about 42 per cent moisture and have a heat value of about 9,000 B.t.u. per pound on the dry basis.

[1] Gottlieb

[2] Dry basis, except the moisture values

Wood and other fibrous fuels are analyzed similar to coal. The ultimate analyses, on the dry basis, of representative fibrous fuels are given in Table 3-6.

40. Heat Value of Fuels.—The *heat value, heating value* or *calorific value* of a fuel indicates its power to produce heat under ideal conditions. In engineering practice in this country, the heat value is given in terms of B.t.u. per pound for solid and liquid fuels, B.t.u. per cubic foot (at 68°F. and 14.7 lb. pressure) for gases. It occasionally happens, however, that the heat value for gases is given at 60 or 62°F.

The hydrogen in fuels burns to form water which is vaporized and superheated to the ordinary flue-gas temperature in the furnace or exhaust-gas temperature in the cylinder of an internal-combustion engine. The heat thus required is lost, giving the fuel what is called a *lower heating value.*[1] The *higher heating value* disregards the loss of heat, above the initial temperature, and is the one most generally used in steam-plant practice.

For most accurate results, the heat values are best determined by means of a fuel calorimeter. Empirical formulas, based on the analyses, are often used for obtaining approximate values.

41. Bomb Calorimeter.—The bomb calorimeter is used almost exclusively for determining the heat values of solid and liquid fuels. The principle embodied in this apparatus is to actually burn a small (1 g.) sample of the fuel in an atmosphere of oxygen under pressure (300 to 350 lb. per square inch) and to measure the heat evolved by the rise in temperature of a definite quantity of water, which surrounds the combustion chamber or bomb. The fuel charge is ignited by an electric circuit, and the temperature rise of the water is read on an accurate and finely calibrated (0.01 to 0.05°C.) mercury thermometer.

FIG 10.—Emerson bomb calorimeter.

The Emerson calorimeter, illustrated in Fig. 10, is a typical and commonly used apparatus for determining the heat value of solid and liquid fuels. It consists, essentially, of a bomb, calorimeter bucket, thermometer, stirring device and an outer insulating jacket.

[1] Lower heating value of hydrogen is approximately 53,100 B.t.u. per pound.

The bomb is made of steel, is lined with non-corrosive sheet metal, and is capable of withstanding high pressures. The upper and lower portions fit together with a lead gasket joint which is held tight by a large nut, as shown.

A fuel sample is carefully prepared and placed in the ignition pan, after which the bomb is assembled and connected to a tank of oxygen. It is filled to a pressure of approximately 300 lb. per square inch, as indicated by a gage in the connection piping. The bomb is then placed in the calorimeter bucket which contains about 1,600 g. of water at a temperature slightly below that of the room air.

After inserting the thermometer and stirring device, and conditions have become constant, the fuel is ignited by an electric circuit. Periodic thermometer readings are taken in order to obtain the maximum. The product of the rise in temperature and the weight of water (plus the water equivalent of the bomb and other immersed metal parts) gives, directly, the heat units emitted, provided it is not necessary to make a correction for radiation. The result is changed to B.t.u. per pound by a simple calculation.

42. Dulong's Formula.—If the ultimate analysis of the fuel is available, a close approximation of the heat value may be obtained by using Dulong's formula. It is based on the assumption that the total heat of the fuel is the sum of the heats produced by the individual combustible elements (C, H and S), which are contained in it, and that all of the oxygen and the necessary hydrogen are in combination as water and are inert. From Table 3-7 (page 65) 1 lb. of carbon has a heat value of 14,600 B.t.u.; "free" hydrogen, 62,100 B.t.u.; and sulphur, 4,050 B.t.u. The free hydrogen is found by subtracting $\frac{1}{8}$ of the oxygen (O/8) from the hydrogen (H), as this is the portion of the hydrogen that is combined with the oxygen of the fuel. Thus,

$$F = 14{,}600C + 62{,}100\,(H - O/8) + 4{,}050S \qquad (47)$$

in which

F = B.t.u. per pound of fuel as fired or dry, depending on the basis of the analysis,

and C, H, O and S are the proportions, by weight, of carbon, hydrogen, oxygen and sulphur per lb. fuel from the ultimate analysis.

The heat value of any fuel (gas, liquid, or solid) may be calculated in a way similar to that used in Dulong's formula. Table 3-7 (page 65) gives general data for this.

Example 3-2.—The ultimate analysis of a sample of Indiana coal (Example 3-1, page 55) gives, in percentage by weight, C, 65.65; H, 5.87; S, 1.51; O, 18.61. Find the heat value of this coal using Dulong's formula.

Solution.

$$F = 14{,}600C + 62{,}100\left(H - \frac{O}{8}\right) + 4{,}050S$$

$$= 14{,}600 \times 0.6565 + 62{,}100\left(0.0587 - \frac{0.1861}{8}\right) + 4{,}050 \times 0.0151$$

= 11,847 B.t.u. per pound of coal, as fired.

43. Junker's Gas Calorimeter.—The *heat value of gas fuels* is generally determined directly, by means of the Junker's type gas calorimeter. The principle involved in this apparatus is somewhat similar to that of the bomb calorimeter and consists of measuring the heat evolved from burning a definite quantity of gas by the rise in temperature of a definite quantity of water.

Figure 11 shows in general the construction features of the Junker's type calorimeter. It is constructed of thin sheet copper and consists of an inner chamber which contains the gas burner and which is surrounded by a suitable annular space for the escapement of the heated gases. The water-jacket tubes surround this space. Suitable thermometers are used to measure the temperature of the water flowing into and out of the calorimeter.

In operation, the gas is carefully metered to the burner, at constant pressure. The burned gases travel up through the inner chamber, down the annular space and out, at approximately room temperature, through the lower right-hand opening. The water enters, as shown, and leaves at the top, having a rate of flow of about 2 liters per minute. The gas consumed for heating a definite quantity of water is measured by an accurate meter; or the water used per cubic foot of gas may be determined. From these readings the heat content per cubic foot of gas is calculated.

If the volumetric analysis of a gas fuel is known, the total heat value may be determined, approximately, by *calculation*, by summing the proportionate heats of each constituent combustible gas (see Table 3-7).

44. Chemistry of Combustion.—During combustion the combustible elements (C, H and S) of a fuel unite with the oxygen of the air to form molecular compounds which, for the most part, pass off as "waste gases." The heat produced during the process depends, of course, on the heat-producing properties of each element. The products of combustion are formed with mathematical precision and depend on the combining power of each element. The quantities, or weights, involved depend on the atomic weights of the elements, and these are arbitrary numbers representing the relative weights, referring

to hydrogen as 1.008 (originally 1.000). Thus, oxygen has an atomic weight of 16 which means that oxygen is slightly less than 16 times as heavy as hydrogen, under the same conditions (see Table 3-7).

Atoms combine to form molecules, the more stable units, and the relative weight of each molecule is the sum of the atomic weights of

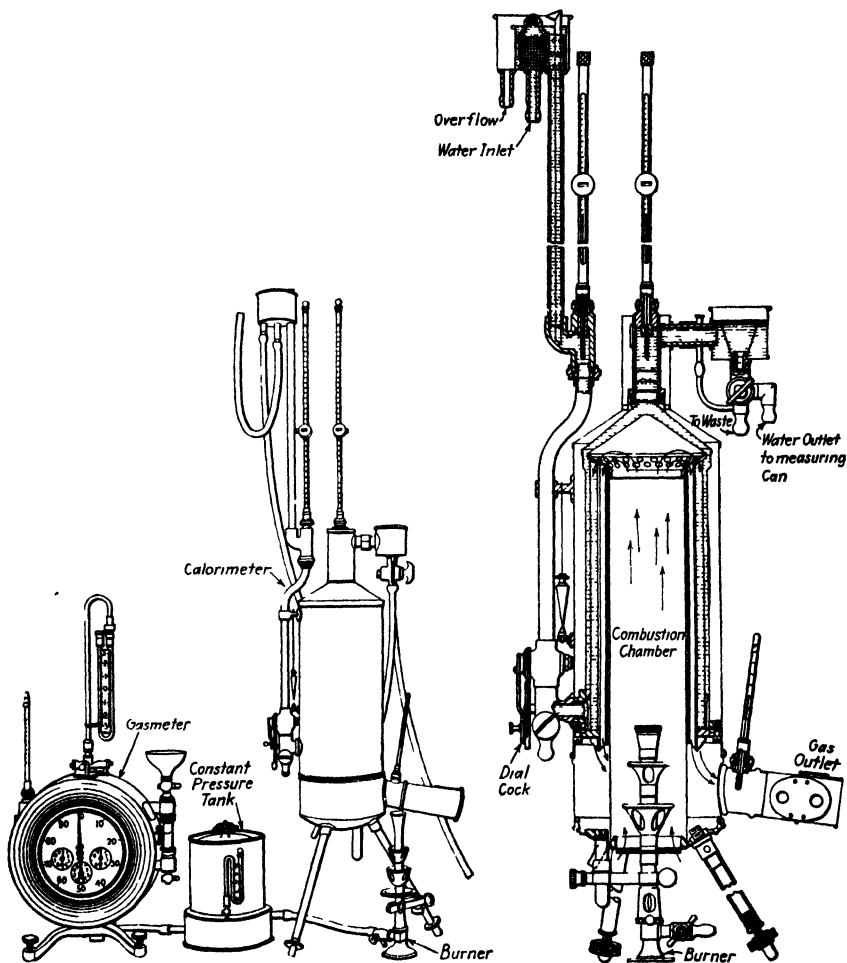



FIG 11 —American Junkers gas calorimeter.

the elements composing it. Thus, in Table 3-7, O_2, H_2, H_2O, etc., indicates that the molecules of oxygen and hydrogen each contain 2 atoms and that their molecular weights are 32 and 2, respectively; also, that each molecule of water (H_2O) contains 2 atoms of hydrogen and 1 of oxygen and has a molecular weight of $2 \times 1 + 16$ or 18.

The carbon in the fuel combines with oxygen to form carbon dioxide (CO_2) or carbon monoxide (CO), depending on the completeness of the combustion. If combustion is complete CO_2 is formed, and the reaction may be written as

$$\begin{aligned} C + O_2 &\rightarrow CO_2 \\ 12 + 32 &= 44 \\ 1 + 2\tfrac{2}{3} &= 3\tfrac{2}{3} \end{aligned} \tag{48}$$

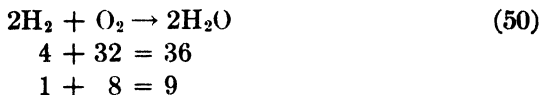
in which the lower figures represent the molecular and combining weights. From this it may be seen that for every pound of carbon burned $2\frac{2}{3}$ lb. of oxygen are required, forming $3\frac{2}{3}$ lb. of the final product, CO_2; and, also, that one volume of carbon combines with one of oxygen to give but one volume of CO_2. Experimental data indicate that approximately 14,600 B.t.u. per pound of carbon burned is liberated during this reaction.

If the combustion is incomplete, carbon monoxide (CO) is formed, and the reaction becomes

$$\begin{aligned} 2C + O_2 &\rightarrow 2CO \\ 24 + 32 &= 56 \\ 1 + 1\tfrac{1}{3} &= 2\tfrac{1}{3} \end{aligned} \tag{49}$$

This shows that for each pound of carbon burned to CO only, $1\frac{1}{3}$ lb. of oxygen are required, and that $2\frac{1}{3}$ lb. of CO are formed; also, that 2 volumes of carbon and one of oxygen produce but 2 of CO. The heat liberated during this reaction is only 4,440 B.t.u. per pound of carbon, indicating a loss of 14,600 − 4,440 or 10,160 B.t.u., due to incomplete combustion.

The volatile combustible matter in solid fuels is composed almost wholly of hydrogen and hydrocarbon compounds such as CH_4, C_2H_2, etc. Hydrogen burns to form water according to the following equation:

$$\begin{aligned} 2H_2 + O_2 &\rightarrow 2H_2O \\ 4 + 32 &= 36 \\ 1 + 8 &= 9 \end{aligned} \tag{50}$$

This equation indicates that 9 lb. of water vapor are produced by the union of 1 lb. of hydrogen and 8 lb. of oxygen, and that 2 volumes of water vapor result from 2 of hydrogen and 1 of oxygen. From the combustion of 1 lb. of hydrogen, approximately 62,100 B.t.u. are liberated, which clearly shows the value of hydrogen to combustion.

The hydrocarbons combine with oxygen to form CO_2 and water vapor. In most cases, the elements C and H are assumed as not being in combination, and calculations are generally made direct from the ultimate analysis.

Sulphur produces a small amount of heat (see Table 3-7) on combining with oxygen and may be assumed to react as follows:

$$\begin{aligned} S + O_2 &\rightarrow SO_2 \\ 32 + 32 &= 64 \\ 1 + 1 &= 2 \end{aligned} \tag{51}$$

which gives the information as mentioned above.

45. Dry Air Theoretically Required for Combustion.—In practice, air is the chief supporter of combustion and may be assumed to be a mixture of oxygen and nitrogen only, in the following proportions:

Element	By volume per cent	By weight per cent
Oxygen	20.91	23.15
Nitrogen	79.09	76.85
	100.00	100.00

Nitrogen is an inert gas and serves only as a dilutent for the oxygen.

From the above proportions it will be seen that each pound of air contains 0.2315 lb. of oxygen and 0.7685 lb. of nitrogen. Thus, for each pound of oxygen, 1/0.2315 or 4.32 lb. of air must be supplied, carrying with it 4.32 − 1 or 3.32 lb. of nitrogen. Also, 1/0.2091 or 4.78 cu. ft. of air are required to produce 1 cu. ft. of oxygen. Each cubic foot of oxygen in the air is accompanied by 3.78 cu. ft. of nitrogen.

In the combustion of each pound of carbon it is seen, from Eq. (48), that 2.67 lb. of oxygen are required. This means that 2.67 (or $2\frac{2}{3}$) × 4.32 or 11.52 lb. of air must be supplied for the complete and theoretical combustion of 1 lb. of pure carbon.

Likewise, from Eq. (50) it can be shown that 8 × 4.32 or 34.56 lb. of air are required for the complete combustion of each pound of free hydrogen. For sulphur, but 1 × 4.32 or 4.32 lb. of air are required.

If the chemical analysis of the fuel is available, the theoretical air required for complete combustion may be calculated from the following equation:

$$w_{at} = 11.52C + 34.56\left(H - \frac{O}{8}\right) + 4.32S \tag{52}$$

in which

w_{at} = lb. of air per pound of fuel.

TABLE 3-7.—CHARACTERISTIC DATA FOR COMBUSTION ELEMENTS

Combustible	Form	Molecular symbol	Atomic weight, approximate	Molecular weight, approximate	Pounds per pound of combustible							Heat value, B.t.u.		Ignition temperature, °F.
					Theoretical required for combustion		Products of combustion in air					Per pound	Per cu. ft. at 68°F. and atmos. press.	
					O_2	Air	CO_2	H_2O	N_2	CO	SO_2			
Carbon (to CO_2)	Solid	C*	12	12	2.667	11.52	3.667	....	8.85	....	....	14,600		
Carbon (to CO)	Solid	C	12	12	1.333	5.76	.....	....	4.43	2.33	....	4,440		
Hydrogen	Gas	H_2	1	2	8.000	34.56	.....	9.00	26.56	....	....	62,100	325	1,130
Sulphur (to SO_2)	Solid	S_2	32	64	1.000	4.32	.....	....	3.32	....	2.00	4,050	......	470
Sulphur (to SO_3)	Solid	S_2	32	64	.....	.....	.....	....	.....	....	....	5,940		
Carbon monoxide	Gas	CO	..	28	0.572	2.46	1.57	....	1.89	....	....	4,380	318	1,210
Methane	Gas	CH_4	..	16	4.000	17.28	2.75	2.25	13.28	....	....	23,850	998	1,202
Acetylene	Gas	C_2H_2	..	26	3.071	13.29	3.39	0.69	10.21	....	....	21,600	1,460	900
Ethylene	Gas	C_2H_4	..	28	3.429	14.81	3.14	1.29	11.38	....	....	21,600	1,570	1,022
Ethane	Gas	C_2H_6	..	30	3.733	16.13	2.93	1.80	12.40	....	....	22,230	1,735	1,000
Hydrogen sulphide	Gas	H_2S	..	34	1.412	6.10	.....	0.53	4.69	....	1.88			
Nitrogen (atmospheric)	Gas	N_2	14	28										
Oxygen	Gas	O_2	16	32										
Carbon dioxide	Gas	CO_2	..	44										
Grain alcohol	Vapor	C_2H_6O	..	46	.....	9.03	1.91	1.17	6.95	....	....	......	1,529	
Gasoline	Vapor	C_8H_{18}	..	114	.....	15.19	3.09	1.42	11.68	....	....	......	5,572	
Kerosene	Vapor	$C_{12}H_{26}$	..	170	.....	15.07	3.11	1.38	11.58	....	....	......	9,011	

* The molecular weight of carbon has not been definitely determined. The latest investigations indicate that a molecule of carbon in any form consists of at least 12 atoms.

C, H, O and S are the proportions, by weight, of carbon, hydrogen, oxygen and sulphur per lb. fuel. $\left(H - \frac{O}{8}\right)$ gives the free hydrogen.

With given conditions of pressure and temperature, the volume of a specified weight of gas or gas mixture may be calculated. From a study of Art. 19 (page 30), it may be seen that the volume of a mol under like conditions of pressure and temperature may be obtained. It follows, then, that the desired volume is in the same proportion to the mol volume as the specified weight is to the mol weight. Thus,

$$PV_m = MRT = 1{,}544T \qquad \text{(for 1 mol of gas)}$$
$$PV = wRT \qquad \text{(for } w \text{ lb. of gas)}$$

Dividing the second equation by the first, the terms P, R, and T, which are the same in both equations, cancel out. This gives

$$V = V_m \frac{w}{M} \tag{53}$$

in which

V = volume of w lb. of gas or gas mixture, cu. ft.
M = weight of 1 mol of gas or gas mixture, lb.
w = total weight of gas or gas mixture, lb.
V_m = volume of 1 mol of gas or gas mixture at given conditions of P and T, cu. ft. Value of V_m, at 14.7 lb. per square inch absolute and 68°F. is 385 cu. ft.

For this purpose, the molecular weight of air may be taken as 28.94. Other molecular weights are given in Table 3-7.

Example 3-3.—It is required to find the weight and volume (at 68°F. and 14.7 lb.) of air required per pound of coal in Example 3-1 (page 55). The ultimate analysis gives, in percentage by weight, C, 65.65; H, 5.87; O, 18.61; S, 1.51.

Solution.—Substituting in Eq. (52)

$$w_{at} = 11.52 \times 0.6565 + 34.56\left(0.0587 - \frac{0.186}{8}\right) + 4.32 \times 0.015 = 8.85 \text{ lb.}$$

$$V_m = \frac{1{,}544 \times 528}{14.7 \times 144} = 385$$

From Eq. (53),

$$V = \frac{385}{28.94} \times 8.85$$
$$= 118 \text{ cu. ft., at 68°F. and 14.7 lb. per square inch absolute pressure.}$$

It should be noted that the volume of air at any temperature or pressure may be found by multiplying the weight by the respective specific volume.

In the boiler furnace, the actual weight of air required for combustion is generally 20 to 100 per cent greater than that theoretically required. This results from the difficulty in mixing the air and combustible under practical conditions.

46. Flue-gas Analysis.—The flue gases or products of combustion resulting from the complete oxidation of a fuel with the theoretical air supply, as previously learned, are N_2, CO_2, water vapor and possibly some SO_2. When the air supply is deficient, CO, H_2 and some of the hydrocarbons are present in the flue gas, indicating that the combustion is incomplete. Since it is necessary to supply an excess of air to insure complete combustion in the practical boiler furnace, it

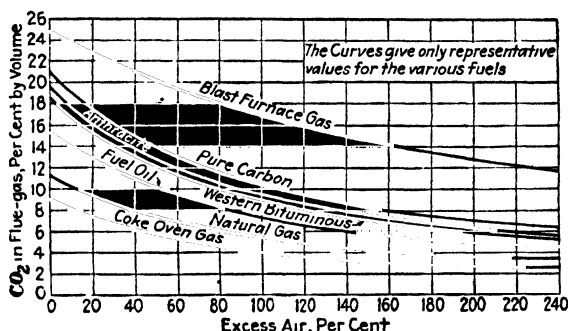



FIG. 12—Curves showing the relation between the CO_2 content in the dry flue gases and excess air, for the complete combustion of typical fuels

is obvious that the oxygen of the surplus air will be discharged to waste and appear in the flue gases.

With a definite quantity of air, the various proportions of the waste gases, N_2, O_2, CO_2, etc., depend on the fuel used as well as on the completeness of combustion. The reason for this lies in the fact that most fuels contain varying amounts of N_2, CO_2, etc., which are liberated during combustion. Thus, in order to determine the amount of excess air and the efficiency of combustion by a study of the flue gases, it is necessary to know the characteristics of the particular fuel used.

Figure 12 shows the relation between the volume of CO_2 in the dry flue gas and the excess air supplied, for complete combustion of various fuels. From this it will be seen that for zero excess air (theoretical air) the CO_2 content in the flue gases ranges between 19 and 21 per cent, for coal and pure carbon. Under actual conditions, with an air excess of 20 to 60 per cent, it will be seen that efficient combustion of most coals may be indicated by from 12 to 16 per cent of CO_2 in the flue gases. The CO_2 content then, is, in general, a significant

indicator of efficient combustion, and apparatus which gives a continuous record of the percentage of CO_2 in the flue gas is, therefore, very desirable. In large plants only, however, is such equipment warranted. A portable, hand-operated apparatus is very frequently used for intermittent and test use.

The presence of CO, in the waste gases, which may be accompanied by a high percentage of CO_2, readily indicates a deficient air supply resulting in incomplete combustion. To obtain a check on the furnace combustion and to get data for the determination of heat losses, it is customary to make a more or less complete analysis of the flue gases. This is accomplished with the *Orsat apparatus*, by means of which the percentages, by volume, of CO_2, O_2, CO and N_2 in the flue gas are determined. The general procedure involved is to pass a definite volume (100 parts) of flue gas through a series of absorbing reagents, each of which absorbs one of the gases, in the order, (1) CO_2, (2) O_2, and (3) CO. The volume, following the absorption of each gas, is carefully measured, giving directly the desired results on a percentage basis. The N_2 is determined by difference, assuming the residual gas to be nitrogen.

In preparation for the analysis it is necessary to obtain a representative sample of flue gas from some point in the boiler setting. The sample should be obtained with a suitable porcelain or steel sampling tube, and care should be taken to obtain the sample over as large an area as possible to avoid the effect due to stratification of the gases. A water aspirator is usually used for extracting the sample, which may be collected in bottles over water. In case a rubber aspirator bulb is used, the sample is pumped directly into the Orsat apparatus.

47. Orsat Flue-gas Analyzer.—Figure 13 illustrates the construction of a well-known portable Orsat apparatus, which is fundamentally the same as all others. It consists essentially of a sampling burette, leveling bottle and three reagent containers which connect with the sampling burette through a common passage, as shown. When the apparatus is in use, a rubber bag is attached to the charging tubes of the O_2 and CO absorbers, 2 and 3, to receive the displacement air from the lower chambers. The sampling burette is surrounded by a water jacket, which is for maintaining constant temperature of the gas sample during the measurements.

Each reagent container consists of an upper and lower chamber, as shown. Steel wool is placed in containers 1 and 2 and copper turnings in container 3, which is for the purpose of exposing a large area of wetted surface to the gases. The reagent in 1 is for absorbing the CO_2 and consists of a solution of one part of caustic potash (KOH) dis-

solved in two parts of water. For absorbing O_2, a solution consisting of 5 g. of pyrogallic acid powder dissolved in 100 c.c. of the KOH is used. The CO is absorbed in container 3, using an ammoniacal solution of cuprous chloride. Copper chips are placed in the solution to keep it energized.

To make an analysis with the apparatus, flue gas is first forced into the sampling burette and allowed to bubble through the leveling bottle

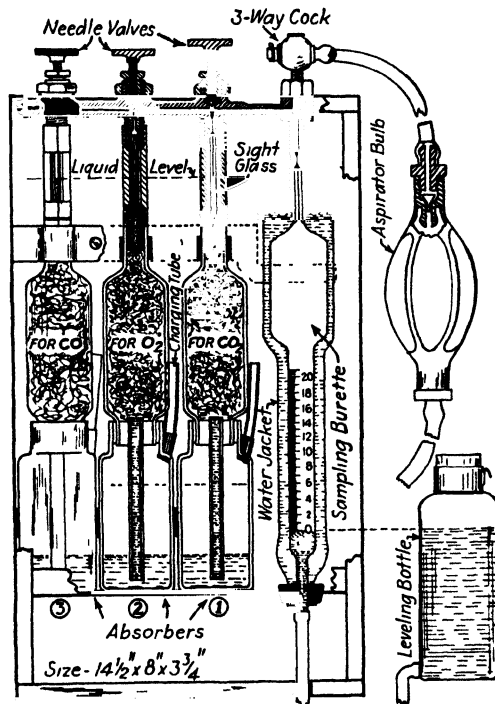



FIG. 13.—Hayes flue-gas analyzer (Orsat apparatus).

which is filled with water. The three-way inlet cock is then turned to allow the surplus gas to pass into the atmosphere as the leveling bottle is brought to the position shown in Fig. 13. At zero the burette contains 100 parts or volumes of gas at the temperature of the water in the surrounding jacket and at atmospheric pressure. The cock is next closed completely, thus trapping the sample to be analyzed. The CO_2 is absorbed out first, and this is accomplished by raising the leveling bottle and opening the needle valve leading into container 1. The reagent flows into the lower part of the container and the CO_2 is absorbed from the sample by its contact with the KOH on the steel

wool. The unabsorbed gas is then drawn back into the burette by a reverse operation, and care should be taken to bring the reagent to its original level. The water in the leveling bottle is next equalized with that in the burette, and the reading is taken from the graduated scale. This gives the CO_2 content. For the O_2 and CO, the operation is repeated, successively, in containers 2 and 3. This requires more time and the gas should be circulated back and forth until a constant reading is obtained. In the manipulation of the apparatus care should be taken not to mix the reagents. A small amount of practice will enable one to make a complete analysis in from 2 to 3 min.

48. Weight of Dry Flue Gas.—By using the Orsat apparatus properly and taking samples of gas from various points in the gas flow, a volumetric analysis of the dry flue gas may be determined which will be truly representative. The results of such an analysis of a dry flue-gas sample may be expressed as an equation. Thus:

$$CO_2 + O_2 + CO + N_2 = 100 \qquad (54)$$

in which

CO_2, O_2, CO, and N_2 = percentages, by volume, of the component gases.

This being on a volumetric basis, instead of percentages by volumes, a convenient unit of volume, the mol or pound-mol, may be used. For example, if the figures 11, 6, 1 and 82, respectively, were assumed to be the percentages of gases in the above equation, it could be written:

$$11 \text{ mols } CO_2 + 6 \text{ mols } O_2 + 1 \text{ mol } CO + 82 \text{ mols } N_2 = 100 \text{ mols}$$

Thus, this can be expressed generally:

$$CO_2 \text{ mols} + O_2 \text{ mols} + CO \text{ mols} + N_2 \text{ mols} = 100 \text{ mols}$$

Then the weight of the 100 mols of flue gas is determined by multiplying the number of mols of each constituent gas by the number of pounds equivalent to each respective molecular weight. This gives the weight of 100 mols as follows:

$$(44CO_2 + 32O_2 + 28CO + 28N_2) = \text{weight of 100 mols} \qquad (55)$$

The carbon of the coal, is apparently the only fuel which burns, as CO_2 and CO are the only new gases formed. The H_2O formed by the combustion of H_2 is condensed and forms liquid water either in the burette or sampling bottle and has no effect on the volumetric relations of the gas sample in the Orsat apparatus. The small volume of SO_2

formed by the combustion of sulphur is absorbed in the apparatus by the KOH solution, thus increasing slightly the CO_2 reading over its true value.

Of the carbon in the coal a small part remains unburned and falls through the grates to the ash pit. The carbon which does burn will occur in the gases of combustion in the CO_2 and CO formed. The portion of carbon in each gas may be figured from the relative combining weight of carbon with oxygen. Hence, the weight (pounds) of carbon which burned to produce the 100 mols of flue gas in Eq. (55) equals

$$\left(\tfrac{12}{44} \times 44CO_2 + \tfrac{12}{28} \times 28CO\right) = 12(CO_2 + CO) \qquad (56)$$

By combining Eqs. (55) and (56), the weight of dry flue gas produced when 1 lb. of carbon burns is determined. Thus:

$$w_{gc} = \frac{44CO_2 + 32O_2 + 28CO + 28N_2}{12(CO_2 + CO)}$$
$$= \frac{11CO_2 + 8O_2 + 7CO + 7N_2}{3(CO_2 + CO)}$$

To simplify, both sides of Eq. (54) are multiplied by 7.

$$7CO + 7N_2 = 700 - 7CO_2 - 7O_2$$

hence

$$w_{gc} = \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)} \qquad (57)$$

in which

w_{gc} = pounds of flue gas formed for each pound of carbon burned.

Then the weight of carbon burned, per pound of coal fired, is determined by the difference between the total weight of carbon and the weight of unburned carbon, thus:

$$C = C_f - \frac{W_aC_a}{W_f}$$
$$C = \frac{W_fC_f - W_aC_a}{W_f} \qquad (58)$$

in which

C = weight carbon burned per pound of coal.
W_f = weight coal burned, lb. per hour.
W_a = weight ash and refuse resulting from the combustion of W_f, lb. per hour.
C_f = weight carbon in 1 lb. of coal (ultimate analysis), lb.

C_a = weight unburned carbon in 1 lb. of ash and refuse, lb.

$$w_g = \frac{W_f C_f - W_a C_a}{W_f} \times \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)} \tag{59}$$

in which

CO_2, O_2, and CO are to be expressed as percentages by volume, from flue-gas analysis.

Other terms as in Eq. (58).

49. Weight of Dry Air Supplied for Combustion.—The weight of air supplied for combustion is necessarily in excess of that theoretically required. The weight of air supplied can be measured, but generally it is determined by means of the flue-gas analysis. The weight of nitrogen per pound of carbon burned is, from dry gas weight

$$w_{N_2c} = \frac{7N_2}{3(CO_2 + CO)} \tag{60}$$

It may be assumed, without serious error, that this weight of nitrogen includes only the nitrogen of the air supplied. From Art. 45 (page 64), air, by weight, contains 76.85 per cent nitrogen. Hence the weight of air supplied per pound of carbon burned is expressed as follows:

$$w_{ac} = \frac{7N_2}{0.7685 \times 3(CO_2 + CO)}$$

The weight of air supplied per pound of coal fired may be expressed as

$$w_a = \frac{W_f C_f - W_a C_a}{W_f} \times \frac{3.03N_2}{CO_2 + CO} \tag{61}$$

in which the symbols are as in Eqs. (54) and (58).

Knowing the weight of air supplied and that theoretically required, the percentage of excess air is calculated from the equation

$$E_a = \frac{w_a - w_{at}}{w_{at}} \times 100 \tag{62}$$

in which

E_a = per cent of excess air supplied for combustion.

w_a and w_{at} = actual and theoretical weights of air per pound of coal, respectively.

Example 3-4.—Using the data from Examples 3-1 (page 55), 3-2 (page 60), 3-3 (page 66) and the following, calculate (*a*) the weight of dry air supplied per pound of coal fired, (*b*) percentage of excess air and (*c*) the weight of dry flue gas per pound of coal.

Coal fired per hour	4,540 lb.
Ash and refuse per hour	380 lb.
Combustible in refuse, per cent	11.2

Flue-gas analysis gave in percentage by volume: CO_2 13.19, O_2 5.96, CO 0.28 and N_2, by difference, 80.57.

Solution.—*a*. Using Eq. (61),

$$w_a = \frac{3.03N_2}{CO_2 + CO} \times \frac{W_fC_f - W_aC_a}{W_f}$$
$$= \frac{3.03 \times 80,57}{13.19 + 0.28} \times \frac{4,540 \times 0.6565 - 380 \times 0.112}{4,540}$$
$$= 11.72 \text{ lb.}$$

b. By substituting the weight of air theoretically required, 8.85 lb. (from Example 3-3), and the weight actually supplied, 11.72 lb. in Eq. (62), the percentage of excess air is obtained.

$$E_a = \frac{w_a - w_{at}}{w_{at}} \times 100$$
$$= \frac{11.72 - 8.85}{8.85} \times 100$$
$$= 32.4 \text{ per cent}$$

c. Using Eq. (59), the weight of dry flue gas per pound of coal is found.

$$w_g = \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)} \times \frac{W_fC_f - W_aC_a}{W_f}$$
$$= \frac{4 \times 13.19 + 5.96 + 700}{3(13.19 + 0.28)} \times \frac{4.540 \times 0.6565 - 380 \times 0.112}{4,540}$$
$$= 12.14 \text{ lb.}$$

Example 3-5.—The fuel of Example 3-1 (page 55) is completely burned with 50 per cent excess air. Determine the volumetric analysis of the dry products of combustion.

Solution.—Combustible elements in 1 lb. of the fuel: C, 0.6565 lb.; "free" H, 0.0354 lb.; S, 0.0151 lb.

Dry products of combustion per pound of fuel (using data from Table 3-7, page 65):

$$CO_2 = 0.6565 \times 3.667 = \qquad 2.41 \text{ lb.}$$
$$SO_2 = 0.0151 \times 2.00 = \qquad 0\ 03$$
$$N_2 = 0.6565 \times 8.85 = 5.81$$
$$0.0354 \times 26.56 = 0.94$$
$$0.0151 \times 3.32 = 0.05$$

$$N_2 \text{ in theoretical air} = 6.80$$
$$N_2 \text{ in excess air} = 3.40$$

$$\text{Total } N_2 = \qquad 10.20$$
$$O_2 \text{ in excess air} = \frac{3.40}{3.32} = \qquad 1.02 \text{ lb.}$$

Dry products of combustion, volumetric analysis:

$$CO_2 = \frac{385}{44} \times 2.41 = 21.10 \text{ cu. ft.} = 12.2 \text{ per cent}$$
$$SO_2 = \frac{385}{64} \times 0.03 = 0.18$$
$$N_2 = \frac{385}{28} \times 10.20 = 140.00 \qquad = 80.7$$
$$O_2 = \frac{385}{32} \times 1.02 = 12.28 \qquad = 7.1$$

$$\text{Total} \qquad = 173.56 \text{ cu. ft.} = 100.0 \text{ per cent}$$

Problems

1. Change the ultimate analysis of Vigo County, Ind., coal given in Table 3-3 (page 53) to the as-received or as-fired basis, giving the moisture as a separate item. Also change to dry basis.

2. Solve as in Problem 1, for Perry County, Ill., coal. Also change to combustible or moisture and ash-free basis.

3. Calculate the heat value, B.t.u. per pound, of the coal, as fired, from Problem 1. Compare with calorimeter value given in the table.

4. Calculate the heat value of the Montana lignite (Table 3-3), on the dry basis. Compare with value from the table.

5. Using the analysis for Tazewell County, Va., coal, calculate (*a*) the heat value, on as-fired basis, and (*b*) the theoretical weight of air required per pound of coal, as fired.

6. Determine the theoretical weight of air required per pound of dry El Paso County, Colo., coal.

7. Solve Problem 6 for Knox County, Ind., coal, as fired.

8. Using the analysis for Pennsylvania fuel oil, Table 3-4 (page 57), calculate its approximate heat value. Compare with given value.

9. If the West Virginia fuel oil (Table 3-4) is completely burned with 30 per cent excess air, calculate the percentage by volume of CO_2 in the dry products of combustion.

10. Calculate the approximate heat value from the analysis of the blast-furnace gas given in Table 3-5 (page 58). Compare with value given.

11. Calculate, using Table 3-7 (page 65), the approximate heat value for the natural gas given in Table 3-5. Compare with value given.

12. Calculate the approximate heat value for the fir wood fuel given in Table 3-6 (page 58). Also determine the theoretical air required per pound for complete combustion.

13. Columbiana County, Ohio, coal was burned with [illegible]0 per cent excess air. Determine the probable flue-gas analysis values in percentage by volume.

14. A blast-furnace gas gives, in the analysis, in percentage by weight: CO_2 17, CO 24, H_2 0.2, CH_4 0.8, and N_2 58. Assuming standard conditions of temperature (68°F.) and pressure, determine the volumetric analysis.

15. During a power-plant boiler test, coal from Letcher County, Ky., was used. A portion of the data taken gave

Coal fired per hour	6,850 lb.
Ash and refuse per hour	313 lb.
Combustible in ash and refuse, per cent	31.5

Flue-gas analysis, percentage, by volume: CO_2 14, O_2 5.5, CO 0.42.

Determine: (*a*) Theoretical weight of dry air required per pound of coal as fired. (*b*) Actual weight of dry air supplied per pound of coal as fired. (*c*) Percentage of excess air. (*d*) Weight of dry flue gas per pound of coal as fired.

16. The following data were taken from a report of a test on a large central power station:
Ultimate analysis, as fired, of coal burned, percentage: S 1.86, H 4.70, C 77.66, N_2 1.45, O_2 5.63, ash 8.70 and moisture from proximate analysis, 2.94.
Flue-gas analysis, percentage by volume: CO_2 13.1, O_2 5.9, CO 0.2.

Coal fired per hour	21,400 lb.
Ash and refuse per hour	2,080 lb
Unburned carbon in ash, per cent	22

Determine the following items per pound of coal as fired: (*a*) Heat value. (*b*) Theoretical weight of air required. (*c*) Actual weight of air supplied. (*d*) Weight of dry flue gas.

17. Solve Problem 16 using the analysis for Morgan County, Tenn., coal. Flue-gas analysis: CO_2 13.4, O_2 5.7, CO 0.1 per cent.

18. A blast-furnace gas at 14.7 lb. and 68°F. has the following analysis by volume: H_2 3.5, CO 30.5, CO_2 10.0, N_2 remainder. Change this analysis to a weight basis.

19. Calculate the weight of air required per cubic foot of the gas of Problem 18.

20. Calculate the volumetric analysis of the wet flue gas from the complete combustion of the gas of Problem 18.

21. Calculate the heating value, B.t.u. per pound of the gas of Problem 18.

22. Using the Tazewell County, Va., coal, Table 3-3, calculate (*a*) ultimate analysis on dry basis; (*b*) flue-gas analysis by volume if the coal is completely burned with 50 per cent excess air.

23. A gas has the following volumetric analysis: H_2 2.2, CH_4 92.5, C_2H_2 1.4, CO 0.8, CO_2 0.5, N_2 2.6. Calculate (*a*) heating value, B.t.u. per cubic foot; (*b*) weight of theoretical air for combustion, lb. per cubic foot of gas; (*c*) analysis of gas on weight basis.

24. Calculate ultimate analysis of the California coal, Table 3-3, on the dry basis.

25. Calculate ultimate analysis of the Perry County, Ill., coal, Table 3-3, on the dry basis.

26. Calculate heat value, B.t.u. per pound as fired, of the Knox County, Ind., Table 3-3, coal.

27. Calculate heat value, B.t.u. per pound dry, of the Oklahoma coal, Table 3-3.

28. Calculate theoretical weight of air per pound of Tennessee coal, Table 3-3, as fired.

29. Calculate volumetric analysis of dry products of combustion if 1 lb. of Letcher County, Ky., Table 3-3, coal burns completely with the theoretical weight of air.

30. Calculate volume of air at 14.7 lb. per square inch absolute and 68°F. required in Problem 29.

31. Calculate per cent CO_2 by volume in the dry flue gas if the third California fuel oil, Table 3-4, burns with 25 per cent excess air.

32. Calculate approximate heat value, B.t.u. per cubic foot, of the coke-oven gas, Table 3-5.

33. Calculate volumetric analysis of dry products of combustion, if 1 lb. of Perry County, Ill., Table 3-3, coal is burned with 40 per cent excess air.

CHAPTER IV

STEAM-POWER BOILERS

50. Discussion.—The first successful attempt to generate steam, under pressure, in an inclosed vessel or boiler, for the purpose of doing useful work, was made near the end of the seventeenth century. Following that time the development of the boiler was slow until the use of steam machinery began to demand boilers of larger capacity and for higher pressures. Almost every conceivable shape of the boiler vessel has been tried with varying degrees of success. Thus, the modern designs are the result of over two centuries of improvement, with the principle remaining essentially the same.

As a source of power for industrial and domestic purposes in the United States, steam furnishes about 75 per cent of the total developed, making the steam-power boiler one of the most vital parts of our industrial system. Recent development in boiler practice has kept pace with industrial efficiency, and in many cases it has been decidedly a leading factor. The most recent development, however, has been toward improvement in combustion and in furnace construction. The present tendency in large power installations is toward higher steam temperatures and pressures, the temperature being the principal limiting factor. Steam at pressures of 1,200 to 1,400 lb. per square inch is now used successfully, and pressures of 600 to 700 lb. per square inch at temperatures of 650 to 750°F. are quite common. The smaller units continue to use pressures of 200 to 300 lb. per square inch and superheat of 150 to 250°F.

For the purpose of this book only boilers which generate steam at pressures above 25 lb. per square inch will be considered, as those for lower pressures are used principally in heating plants.

Boilers can be built for use with almost any kind of fuel which has a good heating value. The main distinction, based on the fuel used, lies in the furnace and stoker design rather than in the boiler-shell construction. These items will be considered in a later chapter.

It should be noted that the term "boiler" applies only to the vessel with the function of steam generation. The term "steam-generating unit" has a wider scope and is defined in the A.S.M.E. Test Code for 1936 as a combination of apparatus for producing, furnishing or

recovering heat, together with apparatus for transferring the heat so made available to the fluid being heated or vaporized. The apparatus of the unit may include any or all of the following equipment: boiler, water walls, water floor, water screen, superheater, reheater, economizer, air heater, furnace, and fuel-burning equipment. The economizer and air heater are included only when the heat absorbed in each is returned to the unit.

51. Classification.—Boilers may be classified in a great many different ways, each having reference to a particular common characteristic. The confusion which usually results from a detailed classification may be avoided by considering that all modern boilers use tubes which serve either as a passage for the combustible gases or as a vessel for the boiling water. Thus, boilers may be classed as:

1. Fire-tube boilers.
2. Water-tube boilers.

The plain cylindrical boiler, without tubes, has now become obsolete and will not be considered here.

In the *fire-tube boilers* the hot gases pass over the inner surfaces of the tubes while the outer surface, which may or may not be totally submerged, emits heat to the water or steam contained inside the boiler. There is a large variety of fire-tube boilers. Their principal difference lies in the arrangement of the tubes within the shell, which in many cases is cylindrical. The shell may be vertical or horizontal, and it may or may not include the furnace.

Boilers of this class have a field of use which is almost wholly apart from that of the water-tube types. In the smaller sizes they are made portable and have an application in connection with steam shovels, logging machinery and hoisting engines. For stationary installations and for use on locomotives, fire-tube boilers are built in sizes up to as high as 7,600 conventional boiler hp. In various sizes and special design the fire-tube boiler is well adapted to marine use. The efficiency is generally low, but because of the low first cost many industrial plants continue to use them.

Where the service demands a large amount of evaporation at high pressures (above 150 lb. per square inch) *boilers of the water-tube type* are used exclusively. In the smaller sizes, of 100 to 350 boiler hp., they compare in capacity with the larger fire-tube boilers, excepting those used on locomotives. They can be used for almost any kind of service: in heating plants, in industrial plants and in central power stations. The highest efficiencies are obtained with boilers of the water-tube type, which is due principally to their frequent installations with mechanical stokers.

The respective positions of the drums and tubes, and the shape of the tubes comprise the principal distinguishing features in different makes of water-tube boilers. Several, however, have a type of construction which bears very little likeness to usual designs.

The choice of a boiler for a particular service depends, to a great extent, on many factors, economical and practical, which are out of the range of this book. In the following articles it is attempted to show the engineering features of a representative number of boilers

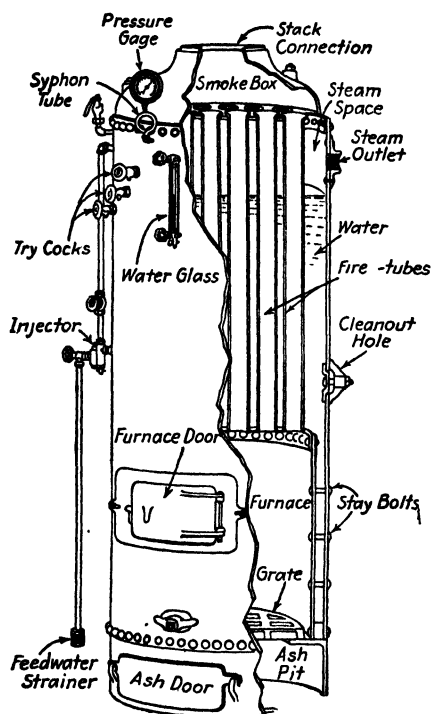



Fig. 14.—Murray vertical fire-tube boiler, with exposed tube sheet.

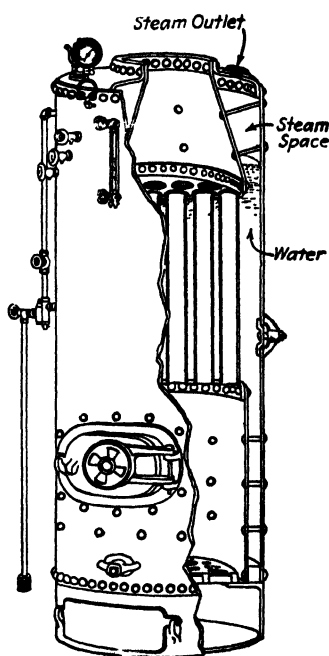



Fig. 15.—Vertical fire-tube boiler, with submerged tube sheet.

commonly used for all types of power service throughout the United States.

52. Vertical Fire-tube Boilers.—Figure 14 illustrates a type of fire-tube boiler that has a wide range of use where compactness, portability, small capacity and low first cost are the main requirements. Boilers of this type are built in sizes ranging from 24 to 72 in. in diameter, and from 60 to 132 in. high, having a heating surface of from 50 to 1,000 sq. ft., which gives a rated capacity of 5 to 100 boiler hp.

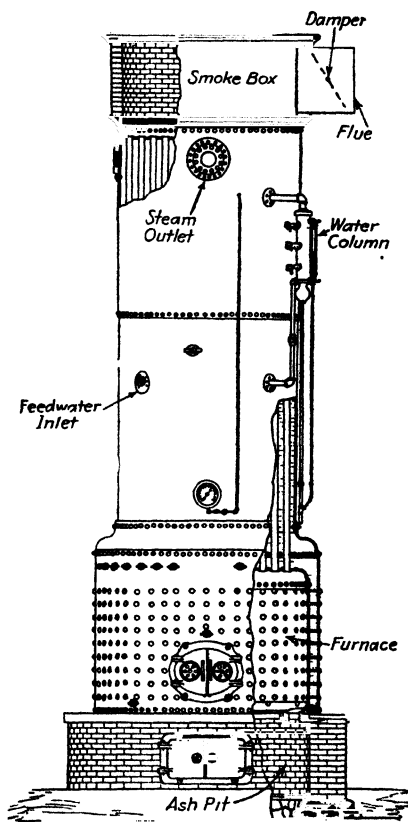
When especially constructed, this type of boiler can be operated continuously at a pressure as high as 225 lb. per square inch.

The fire box and furnace are surrounded by a water leg that is sufficiently stayed to avoid crushing or buckling of the inner plate. Hand holes are provided at points in the shell to enable thorough internal cleaning. The accessories necessary to the operation of the boiler are a water-level gage glass, pressure gage, grates, safety valve, steam connection, a blow-off cock, feed-water piping and an injector. A water column is used with the larger boilers.

The distilled gases, when burning, pass through the tubes into the smoke box and out through the stack to the atmosphere. Under rapid firing the tube sheet (Fig. 14) often becomes overheated, causing considerable leak trouble at the tube joints. This condition is overcome by submerging the tube sheet as shown in Fig. 15. In the *submerged tube-sheet boiler* the steam space surrounds the smoke box, which may result in the production of high-quality or slightly superheated steam.

Fig. 16 – The Manning boiler.

In order to increase the heating surface and combustion space without increasing the number of tubes, the *extended or enlarged fire-box* construction is used. This feature also serves well in the case of burning wood or other bulky fuels. Such a boiler lends itself readily for use on logging equipment and is commonly used in the lumber regions of the Northwest.

The *Manning boiler* (Fig. 16) is built in sizes up to 408 boiler hp., with a maximum height of 26 ft. 10 in. and inside furnace diameter of 9 ft. This boiler gives considerably improved efficiency by virtue of its longer tubes or gas passages. The enlarged and strongly stayed circular fire box rests on a permanent brick or concrete setting which forms the ash chamber. The connection between the fire box and the

upper part of the boiler is made with the "Ogee" ring, as shown in Fig. 16. Frequently, however, a tapered course is used. The gases leave the brick-lined smoke box through a horizontal breeching which connects with the stack.

Vertical fire-tube boilers are very sensitive to firing conditions, and their efficiency is comparatively low, owing to the quick passage of high-temperature gases into the stack. When used with oil as a fuel an improved efficiency is usually obtained. One of the chief objections to vertical fire-tube boilers is the adherence of soot to the tubes, espe-

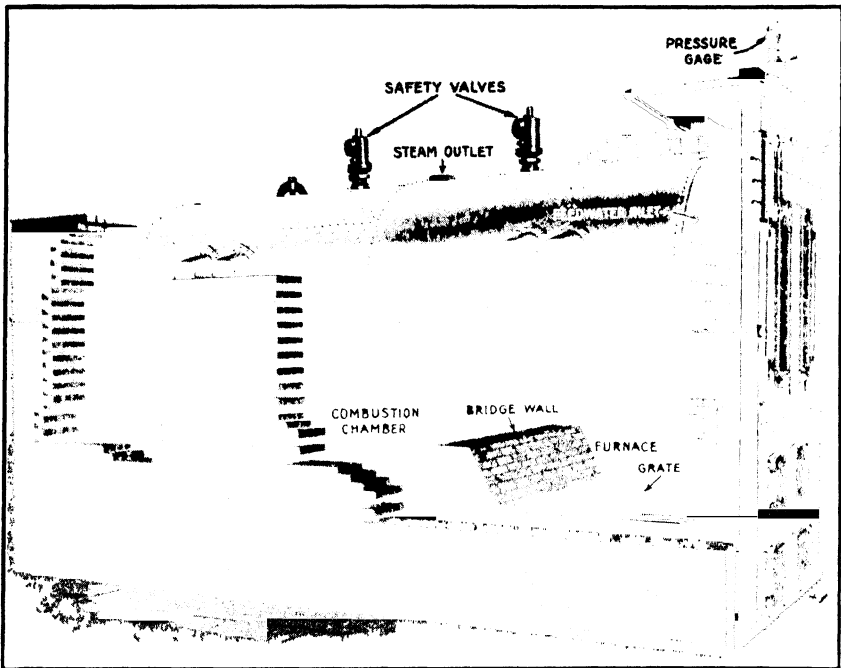



Fig. 17.—Erie City horizontal-return tubular boiler.

cially when used with high volatile coals. They were designed originally for hard and smokeless coals, and when used with soft coal it is recommended that they be built with 3-in. tubes.

53. Horizontal-return Fire-tube Boiler.—Figure 17 shows the characteristic features of a *horizontal-return tubular boiler* ("H.R.T.") and setting. The drum is mounted to provide for expansion by means of side brackets which rest on the brickwork of the side walls or by means of hangers which suspend from column-supported cross-beams near the ends of the drum. When brackets are used, those near the front are permanently fastened, while those in the rear are free to move

on rollers, with the boiler. Suspension from cross-beams is resorted to for the large-sized boilers.

The lower part of the setting comprises the furnace, and this is lined with fire brick. The side walls close in on a level with the axis of the drum, and the rear of the setting is covered by an arch at a height just above the top row of tubes. The fuel chamber can be built for burning any kind of power fuel.

The gases pass from the fuel area along the under surface of the drum to the rear of the setting, thence through the tubes in a reverse direction to the smoke box, past the damper, and, finally, through the breeching to the stack.

The feedwater pipe enters the boiler above the tubes, at the front and side of the drum, and extends toward the rear to over half the

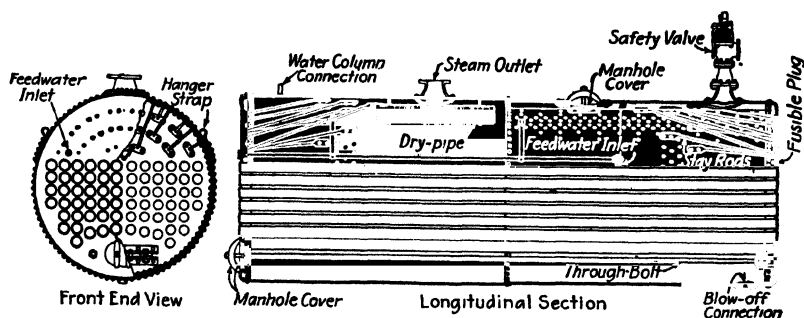



FIG. 18.—Showing shell construction of a horizontal-return tubular boiler.

length where the inlet is provided. In some cases the lower side of the pipe is perforated to allow for discharging over a considerable length. The water circulation is usually considered to be down in the middle of the drum and up at the ends.

The drum (Fig. 18) is sufficiently stayed to prevent bulging of the ends, and at least two manholes for internal cleaning and inspection are provided. Frequent blow-offs remove a portion of the heavy solid impurities which settle at the bottom. To remove the soot from the tubes, the flue doors at the front are opened, and the soot is blown by means of steam or air jets to the rear of the setting from where it is removed by shovels through clean-out doors in the side or rear walls.

Figure 19 is a photograph of a welded drum of a horizontal-return tubular boiler. No rivets are used. The upper part of the head is shaped back to offer resistance to deformation due to internal pressure. No stay rods or braces are used in this design.

Fusion-welded drums are finding wide use in water-tube boilers, especially for high-pressure service. This type of fabrication is rare for fire-tube boilers.

The horizontal-return tubular boiler is built in sizes of 36 in. in diameter and 8 ft. long, to 84 in. in diameter and 20 ft. in length. Capacities range from 15 to 350 hp., with pressures up to 175 lb. They are somewhat more efficient than vertical fire-tube boilers, and their field of use lies principally with small industrial plants where low first cost and simplicity of operation are the main items considered.

54. Down-draft Fire-tube Boiler.—The down-draft, two-pass boiler, shown in Fig. 20, represents a combination of the down-draft furnace and the two-pass construction. The down-draft feature is also used with the type of boiler shown in Fig. 17.

A series of water tubes extending the full width of the furnace form the upper grate (see Fig. 20). These tubes are joined to a front and

Fig. 19.—Fusion-welded, braceless, H.R.T boiler. (*Henry Vogt Machine Company*)

rear header, both of which have their ends connected to the boiler shell below the water line. Coal is fed to the upper grate where distillation takes place. Sufficient air is admitted above the coal bed, which, with the distilled gases, travels downward to the combustion chamber. The burned gases then continue through the lower pass of tubes to the rear chamber, thence in a reverse direction through the return or upper set of tubes to the smoke box at the front of the setting. The lower grate serves to catch and more completely burn any combustible that falls from the upper grate. Boilers equipped with the down-draft feature can be fired with almost smokeless combustion, and for this reason they are particularly adaptable to city use.

The two-pass construction aids in increasing the boiler heating surface and is most frequently used without the down-draft addition. Each of these features assists in more efficient operation of horizontal fire-tube boilers.

55. Locomotive Boilers.—Boilers used on locomotives are of the horizontal fire-tube type and are generally built to operate with high-combustion rates. Their efficiencies are exceptionally low because of the rapid flow of the hot combustion gases through the tubes. Locomotive boilers are built in a large number of sizes, ranging as high as 7,600 sq. ft. in amount of heating surface and 125 ft. in length. The

FIG. 20.— Casey-Hedges down-draft boiler.

larger and modern boilers are generally built with provision for a superheater.

Figure 21 shows the general construction and features of a modern locomotive boiler. The fire box or furnace is surrounded by a water leg or mud ring and is separated from the combustion chamber by a row of water tubes b and the combustion arch b_1. The arch prevents the hot gases from impinging directly on protruding edges and provides for lengthening the gas travel. In the complete assembly of this boiler, the upper or larger fire tubes a contain the superheater tubes which extend from the front end. All flattened and irregular surfaces are stayed to avoid buckling. A steam outlet pipe leads from the steam dome d and extends along the upper part of the steam space, leaving through the opening in the upper portion of the front tube sheet. The fuel may or may not be fed to the furnace through the fire door in

the rear. The hot gases travel through the tubes to the smoke box at the front end, and out, vertically, through a smokestack. The opening at the bottom of the smoke box provides for the insertion of the exhaust nozzle, leading from the engine, which creates the draft for the furnace.

FIG. 21.—Locomotive boiler. (Courtesy of International Text Book Company.)

56. Scotch Boilers.—Scotch boilers are of the horizontal fire-tube type. They represent an approach toward more complete internal firing, the main result of which is increased heating surface. When bulky and high volatile fuels are used with a Scotch boiler, the Dutch-oven or external furnace is employed. In most cases, however, this type of boiler is completely self-contained. Their quick steaming ability and compactness make them particularly adaptable to marine and industrial use, where floor space and head room are limited.

Figure 22 shows, in a general way, the construction of a dry-back Scotch boiler. The furnaces are cylindrical in shape, the plate being corrugated to withstand radial pressure. The free area at the ends of the drum is strongly supported by through bolts.

There is almost no circulation of the boiler water, which results in a "dead" region under the furnaces. The *Brady Scotch boiler* (Fig. 23) shows a tendency to improve the water circulation by the addition of a steam and water drum, and an annular ring inside and at the front of the main shell. The ring, which is open at the top and bottom, serves as a channel for the water and directs the circulation from the upper drum around the sides of the shell to the region beneath the fire

boxes. This does away with the dead region and gives an induced flow as indicated in the figure. The jacketed-back features, as shown in Fig. 23, are also used in other designs of the Scotch type.

Scotch boilers are built in sizes ranging from 6 to 300 hp., the largest being about 120 in. in diameter and 20 ft. in length. The number of furnaces per boiler depends wholly on the size; four are used in the largest.

57. Babcock and Wilcox Water-tube Boiler—Longitudinal Drum. Figure 24 shows the general layout of a horizontal- and longitudinal-drum Babcock and Wilcox boiler, which is one of the most widely

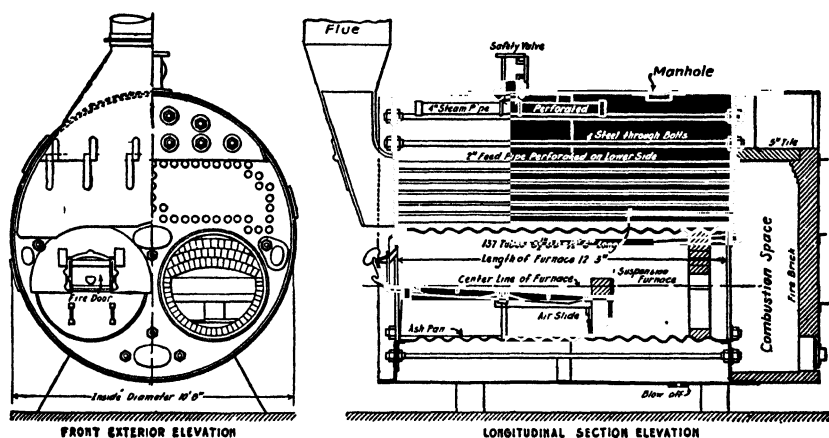



FIG. 22.—Scotch boiler having a dry back.

used water-tube boilers made in the United States. The outstanding feature of this boiler is the sinuous header (Fig. 26), which is constructed to arrange the attached tubes in staggered vertical rows. This arrangement eliminates direct vertical gas passages, causing greater turbulence of the hot gases and resulting in a greater heat absorption per tube. The tubes are expanded into the headers which may be either cast iron or steel, and vertical or inclined. When inclined, the tubes, which are set at an angle of 15 deg., enter the headers at right angles. Vertical headers join with the tubes at an angle, as shown in Figs. 24 and 25.

Handholes (Figs. 26 and 27) are provided opposite the tube holes in each header, for the purpose of assembly and cleaning or replacing tubes. In the vertical headers the handholes are elliptical to permit tube replacement. The surfaces surrounding the holes are smoothly machined to form gasket joints with the handhole covers.

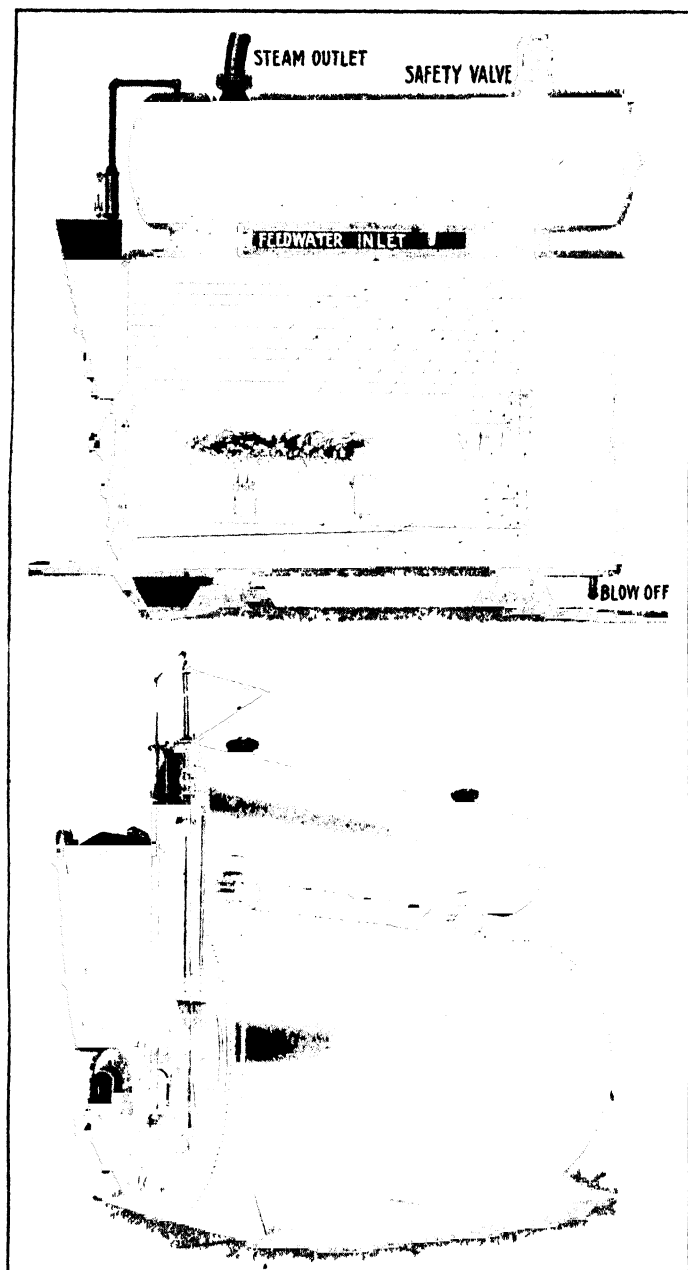



FIG. 23.—Brady Scotch boiler.

A staggered vertical row of tubes composes a tube section. The sections, usually 14 or more in number, are placed together to form the bank of tubes for each boiler Each header meshes snugly with the one adjoining, and usually 7 tube sections are connected to each drum There are usually two drums, ranging in diameters from 36 to 60 in , per boiler Short tube nipples connect the ends of the headers with cross-boxes that are riveted to the under surfaces and near the

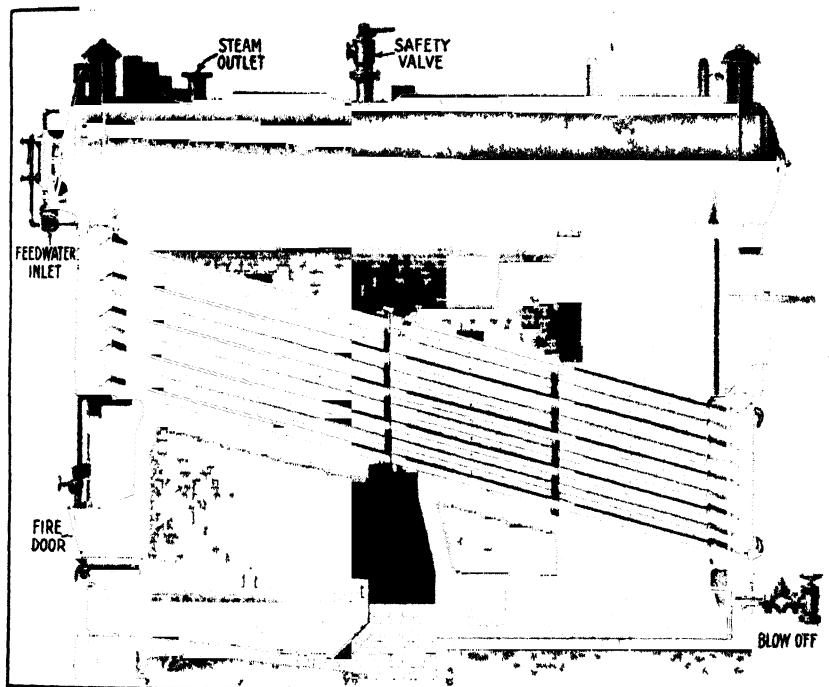



FIG 24 —Babcock and Wilcox longitudinal-drum boiler

ends of the drums The lower ends of the rear headers are connected by tube nipples to a single forged-steel mud box where the major portion of the precipitated impurities settle The blow-off connection makes possible "blowing down" at intervals to remove the settled impurities Manholes at the ends of each drum give access for cleaning and repairing the upper portion of the boiler.

Dryness of the generated steam is effected, partially, by a horizontal dry pipe which connects with the steam outlet and the baffle that is suspended over the path of the steam bubbles which rise from the front headers The main function of the dry pipe, however, is to distribute the steam outlet along the upper region of the drum. The

baffle deflects the bubbles and, to some extent, eliminates the tendency of water particles to emerge into the steam space.

Feedwater is delivered to each drum, at the front and directly into the path of the water circulation, as shown in Fig. 25. The inclination of the tubes, and the difference in density between the water in the rear headers and the hotter water and steam in the front headers, cause the circulation. The gases travel transversely with the tubes, from the hottest region at the front of the setting through the three vertical passes before reaching the breeching at the rear.

The baffles are of cast iron and masonry construction, supported, as shown in Fig. 24, by the tubes and brickwork. The boiler is supported by stirrups from column-supported cross-beams near the ends of the setting. This provides for free expansion and contraction of the boiler without injuring the brickwork of the walls. When more than a single drum is used the saturated steam outlets from individual drums are connected by a cross-pipe, which provides means for union of the steam spaces.

Fig. 25.—Section through front end of Babcock and Wilcox boiler.

Fig. 26.—Vertical steel sinuous header.

All parts of the setting are accessible through door openings in the furnace walls, thus making repair and thorough cleaning possible.

Though Fig. 24 shows a hand-fired setting, most modern installations of water-tube boilers are equipped with a means of supplying the fuel mechanically.

The *Babcock and Wilcox longitudinal-drum boiler* is built in various sizes above 75 hp., for any pressure now used. The number and height of the tube sections, the length of the tubes and the number of drums determine the size or the amount of heating surface per boiler.

58. Heine Boiler.—Figure 28 shows an illustration of a typical *longitudinal-drum water-tube boiler* using the box-header construction. The tubes are expanded into the headers in staggered vertical order, with a handhole opposite each tube connection. By using hollow stay bolts to strengthen the flat surfaces of the header, the rupture of any one bolt will allow water to escape. In this way a break will be readily detected. The holes in the stay bolts also serve as openings for blowing soot and scale from the tubes between the baffles, by means of steam lances. Figure 29 shows the construction of the box header and its connection to the drum. Another distinguishing feature is the location of the mud drum in the main drum of the boiler. The mud drum is merely a pipe about 6 in. in diameter, open at one end and connected with the blow-off pipe at the other. Feedwater is discharged into the open end, and after being heated to the temperature of the boiler water, and after most of the precipitates have been deposited, the direction of flow is reversed, and it enters the boiler proper. Any additional sediment is deposited at the bottom of the rear header and is removed through the drain.

Fig. 27.—Showing handhole and fittings of sinuous header.

It should be noted that the main drum is inclined with the tubes, and that the gases leave the setting in a vertical direction from around the rear end. The boiler is supported by columns in the under setting, the rear water leg being mounted on rollers. Other features, such as water circulation and the steam outlets, etc., in the two boilers are also similar. One manhole at the rear end provides for entrance to the main drum.

The hot gases travel through three passes that are formed by the horizontal baffles. The construction of the baffles is shown in Fig. 30.

Heine boilers, of this type, are built in sizes of from 100 to 1,200 hp. and for pressures up to 500 lb. per square inch. The larger ones have two and three drums attached to the same headers. There are

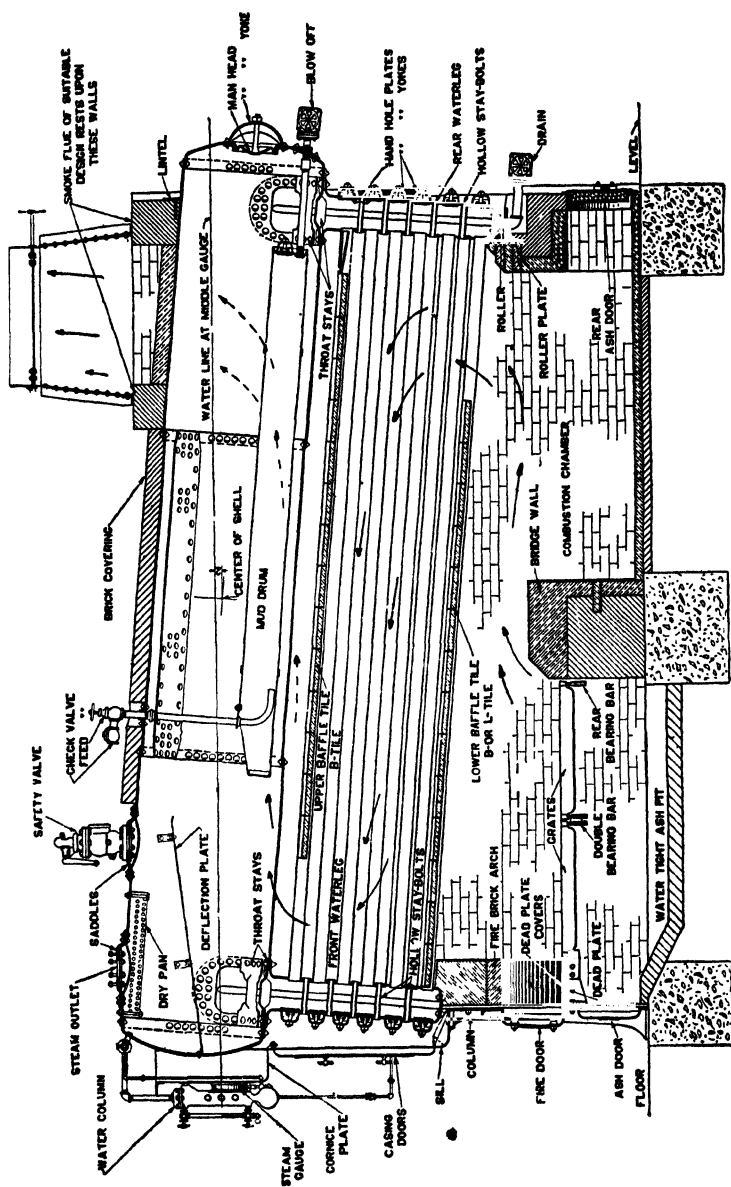



Fig. 28.—Heine boiler.

many manufacturers of longitudinal-drum, water-tube boilers. The two discussed should be taken as typical examples, only.

59. Water-tube Boilers. Cross-drum Type.—Boilers of this type differ from the water-tube boilers described in the preceding articles mainly in the position of the drum, which lies transverse with the tubes. This arrangement allows for a much wider bank of tubes per boiler and facilitates a greater heating area. Figures 31 and 32 show two widely used cross-drum boilers. The water circulation and path of the hot gases are evident on noting the tube and baffle arrangement. The method of mounting, as in the case of all water-tube boilers, is such as to allow free expansion and contraction of the shell

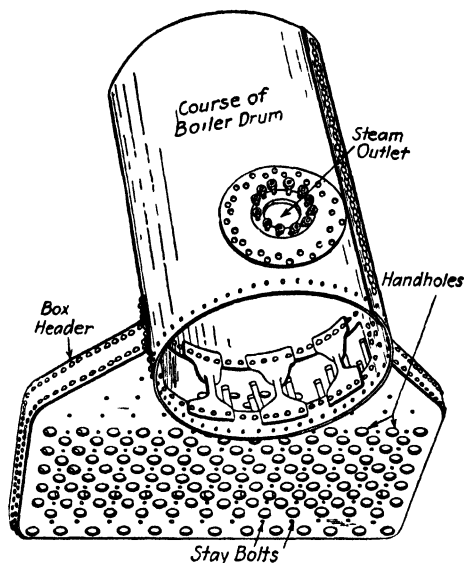



FIG. 29 Showing the box-header construction, Heine boiler

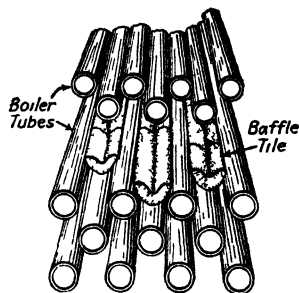



FIG. 30 Horizontal B-type baffle.

Figure 31 illustrates a complete steam-generating unit, the equipment shown being of Babcock and Wilcox manufacture. The boiler is of the sectional-header, cross-drum type, 42 sections wide, 18.5 tube rows high with 4-in. tubes, 24 ft. long, providing a heating surface of 22,235 sq. ft. The drum is 60 in. in diameter, and designed for a pressure of 800 lb. per square inch. Tubes are spaced into two decks, with intervening space occupied by a superheater of the continuous-tube type. In the top of the steam drum is placed a steam scrubber through which the steam passes on its way to the outlet nozzle. In this, steam is washed by the incoming feedwater and its solid concentration thereby reduced. The gas flow is across the tubes, the gas crossing the upper tube deck three times.

The gas leaving the setting under the drum flows downward through a return-bend, straight-tube-type economizer, arranged

24 tubes wide and 24 tubes high. Each tube is 2 in. in diameter and 27 ft. in exposed length, the total heating surface being 8,140 sq. ft. The water flow through the economizer is upward, the top row of tubes entering the boiler drum.

In the last pass of the gas is the 36,540 sq. ft. air heater, composed of 2⅜-in. elements 27 ft. in exposed length (vertical), 34 rows of elements wide and 64 rows deep.

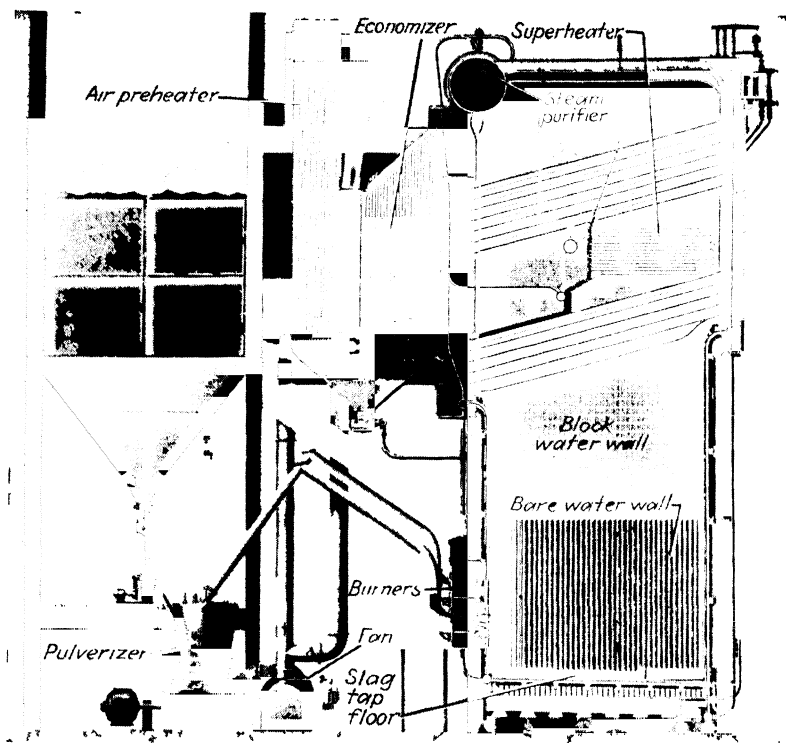



FIG. 31.—Babcock and Wilcox cross-drum boiler.

Three high-capacity pulverizers each with two rows of pulverizing balls prepare the 13,000-B.t.u. West Virginia coal for the boiler, and six cross-tube, pulverized-coal burners burn the coal, directing the burning fuel downward close to the floor. The furnace is the Bailey water-cooled slag-top furnace.

The following table (Table 4-1) gives typical data and results on this steam-generating unit:

The *Springfield boiler* (Fig. 32) uses a sinuous header, differing from the Babcock and Wilcox and other sinuous headers in that the

offsets are by groups of four tubes, as shown in Fig. 33. One handhole accommodates each group of tubes.

60. Water-tube Boilers. Multiple-drum, Bent-tube Type.—Boilers of this class are commonly used for all types of service throughout the steam-power field. They are built with two or more drums, the connections between the drums being by tubes bent at the ends so that

TABLE 4-1.—PERFORMANCE DATA, BABCOCK AND WILCOX CROSS-DRUM UNIT (Fig 31)

Item	Value
Steam, actual, 1,000 lb per hour	335
Liberation, k[1]B per cubic foot per hour	25 0
Solid fuel, coal, 1,000 lb per hour	32 3
Burners, number in use per furnace	4 0
Secondary air pressure in burner casing, inches of water	3 5
Steam pressure at superheater outlet, lb per square inch	730
Pressure drop, drum to superheater outlet, lb per square inch	23
Pressure drop through economizer, lb per square inch	21
Superheated steam temperature, °F.	800
Temperature of flue gas leaving boiler, °F	740
Temperature of flue gas leaving economizer, °F	553
Temperature of flue gas leaving air heater, °F	337
Temperature of water entering economizer, °F	350
Temperature of water leaving economizer, °F	404
Temperature of air entering air heater, °F	80
Temperature of air leaving air heater, °F	343
Draft loss through boiler and superheater, inches of water	3 0
Draft loss through economizer, inches of water	1 25
Draft loss through air heater, inches of water	1 80
Flue gas, 1,000 lb per hour	405
Air through air heater, 1,000 lb per hour	334
Excess air leaving boiler, per cent	22
CO leaving boiler, per cent	0
Heat loss in flue gas (wet gas basis), per cent	10
Heat loss due to unburned carbon, per cent	0 9
Heat loss, radiation and unaccounted for, per cent	3 1
Efficiency of unit, per cent	86 0

[1] kB refers to the unit of 1000 B t u.

they may enter the drums radially. Because of the large number of tubes and the arrangement of the heating surface these boilers have comparatively high capacity.

Figure 34 shows a typical *Stirling boiler* with unit-fired pulverized-coal system. The boiler with 15,136 sq. ft. of heating surface is equipped with the continuous-tube-type superheater, and a tubular-type 20,000-sq.-ft. air heater. The Bailey block-covered furnace includes water-heating surface as follows: side walls 716 sq. ft., down-

take wall 420 sq. ft., uptake wall 435 sq. ft., slag-tap floor 362 sq. ft. The furnace volume is 11,760 cu. ft. Steam leaves the boiler at the rear top drum and goes through the superheater. The first tube bank includes few tubes widely spaced, the function being more as a protection against high temperature for the superheater tubes. The baffling arrangement is such that the first two banks of tubes and the super-

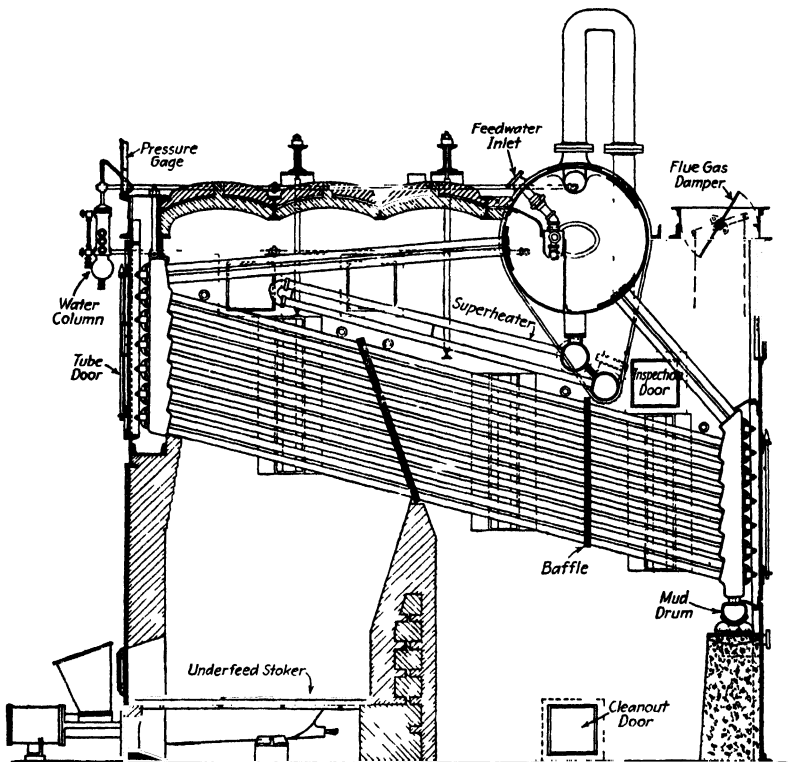



FIG. 32.—Springfield cross-drum boiler, set with an underfeed stoker.

heater receive gas flow mostly across the tubes, and in the rear bank the gas flow is along the tubes.

The *Badenhausen boiler* (Fig. 35) illustrates a similar design which has an unrestricted water circulation, sometimes called "ring-flow." Feedwater is admitted into the lowest drum where it is picked up by the circulating water, which flows in a counter-clockwise direction (Fig. 35) through the three larger drums and the connecting banks of tubes. The smaller drum at the front and top of the setting is a steam-outlet drum only, receiving steam which is superheated as it flows through the top connecting tubes from the upper drum in the

rear. The baffling is quite unusual, causing the hot gases to flow in a somewhat circular path before reaching the last upward pass next to the rear wall.

Fig. 33.—Springfield sinuous header

Figure 36 shows a *Casey-Hedges two-drum boiler*, sometimes classed as a vertical water-tube boiler, having three parallel banks of tubes. The row of tubes between the middle and the front and rear tube banks are called baffle-retaining tubes. The shelf-like baffle extensions in the middle pass give a slightly transverse flow to the hot gases flowing over the middle bank of tubes. Feedwater enters near the side of the upper drum, and circulation takes place down the two rear banks and up the front bank of tubes. This type of boiler is easily forced, and it usually operates at considerably above the rated capacity.

The *Ladd boiler* is similar in design to the Casey-Hedges two-drum boiler. Figure 98

Fig. 34.—Stirling boiler. (*Babcock and Wilcox Company.*)

(page 184) shows a large boiler installation made up of two Ladd boiler units, in duplex arrangement.

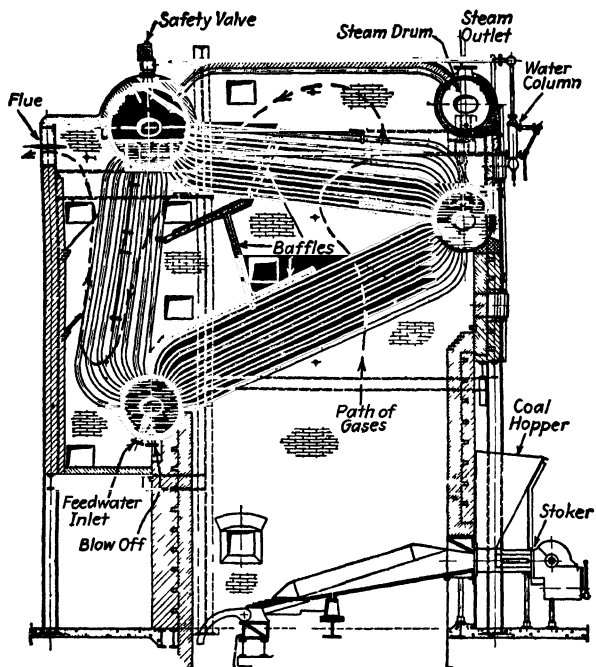



FIG. 35.—Badenhausen boiler, set with an underfeed stoker.

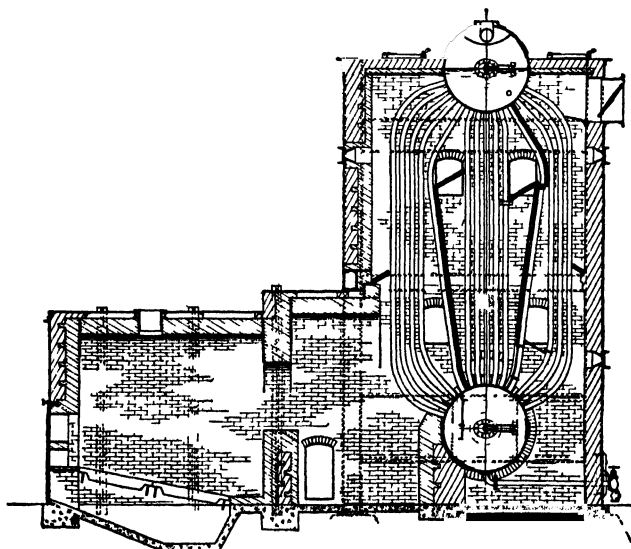



FIG. 36.—Casey-Hedges vertical water-tube boiler, equipped with a "hogged-wood" Dutch-oven furnace.

The chief disadvantage of bent-tube boilers is the necessity of carrying a large stock of replacement tubes, due to their dissimilarity and the large setting generally required.

61. Water-tube Boilers. Miscellaneous Types.—Figure 37 shows a *Wickes vertical boiler*, which consists of an upper and lower cylindrical drum and the connecting vertical straight tubes. To the upper or steam drum are attached the feed pipe, steam outlet and water column.

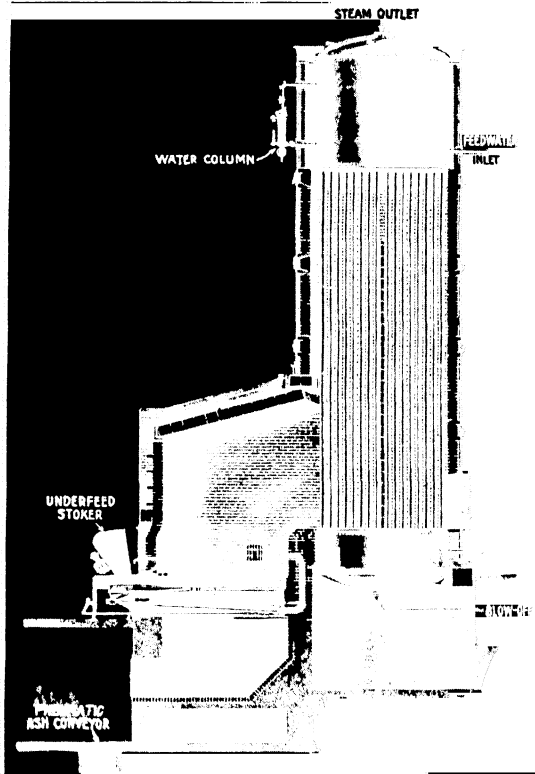



FIG. 37.—Wickes vertical water-tube boiler.

The blow-off is in the lower drum. The water circulation is down the rear and up the front section of tubes. A vertical baffle divides the tubes into two sections and the convection chamber into two passes.

The extended or Dutch-oven furnace is used with this boiler, and the entire setting is enclosed in a steel casing lined with insulating material. Such a casing prevents loss of heat and air infiltration and gives considerable added strength.

This boiler is built in sizes up to 500 hp. and for pressures up to 200 lb. per square inch. It is used with all kinds of fuel and is fre-

quently installed to utilize the waste heat from manufacturing processes, in steel mills, etc.

The *Bigelow-Hornsby boiler* (Fig. 38) is made up of a number of cylindrical units having a common steam and water drum. Each unit consists of an upper and lower drumhead connected by 21 straight tubes, and four units, interconnected by nipples, as shown in Fig. 38,

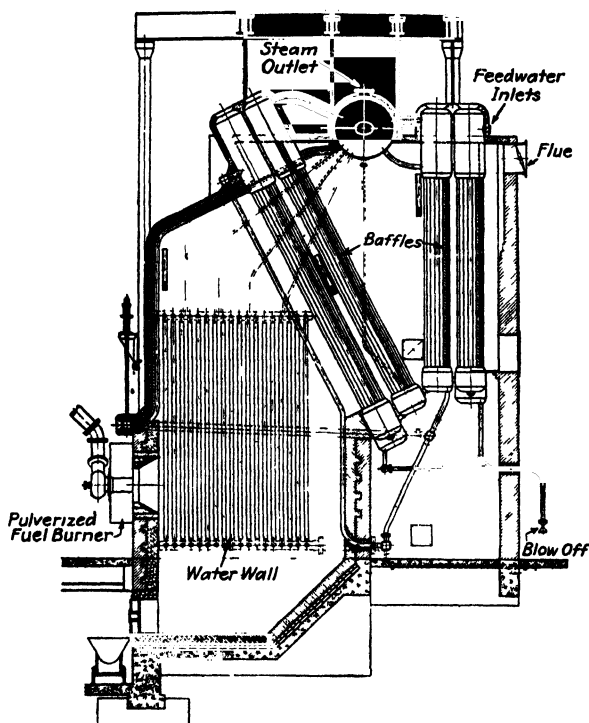



FIG 38—Bigelow-Hornsby boiler with combination front water wall and arch side water walls, and combination water-cooled bridge wall and slag screen Furnace designed for burning pulverized fuel

compose a section. Each boiler consists of three or more sections and a main steam drum and ranges in sizes of from 375 to 1,500 hp

The water circulation is down in the rear and up in the front banks of tubes, to the main steam drum from the front drumheads, and finally to the rear drumheads where the circuit is completed. The feedwater is admitted in the top rear drumheads and mingles with the downward circulating currents. A water level is maintained at a height of about one-third the diameter of the main steam drum. Blow-off connections are made to the lowest parts of the boiler. The products of combustion, after leaving the furnace, flow transversely

over the nests of tubes, in ribbon-like streams, to the outlet at the top of the rear wall. The type of baffle used with this boiler (shown in Fig. 39) produces a high-velocity, turbulent gas flow.

62. Modern Steam Generators.—The boiler unit illustrated in Fig. 40 is an example of one type of modern practice in building steam-generating apparatus for large central stations. The combined unit consists of a relatively small boiler, water walls, superheater, economizer, air preheater, and pulverized coal-burning equipment. The boiler proper consists of an upper steam and water drum, one water drum below, and a small upper dry drum, with headers at the sides and

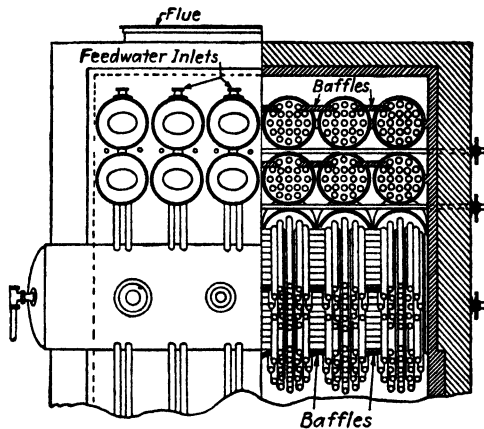



FIG. 39.—Plan of rear portion of Bigelow-Hornsby boiler, showing drum arrangement and method of baffling.

front of the setting, all of which are interconnected by water tubes. The side water-wall tubes are joined to upper and lower headers which in turn connect with the upper and lower drums at the rear. This combination of boiler and side walls gives a nearly cubical furnace that is completely surrounded by radiant-heat absorption surface.

The coal distributor feeds fuel to the burners at the four corners of the setting, and the burners are so arranged that their flames are directed tangent to an imaginary cylinder in the center of the furnace. This gives a spiral motion to the gases as they flow upward through the water screen which cools them below the fusion temperature of the ash. The flow continues through the superheater, down through the convection tubes, and out, near the bottom of the setting, through the air preheater, and, finally, through the induced-draft fan to the stack. The by-pass damper provides for passing a part of the gas away from the superheater. The primary preheated air for the burners flows first to the pulverizing mill from the air heater, and

thence, laden with powdered coal, to the distributor. Ducts at the four corners of the setting carry the preheated secondary or draft air to the burners from the encircling header which leads from the air preheater.

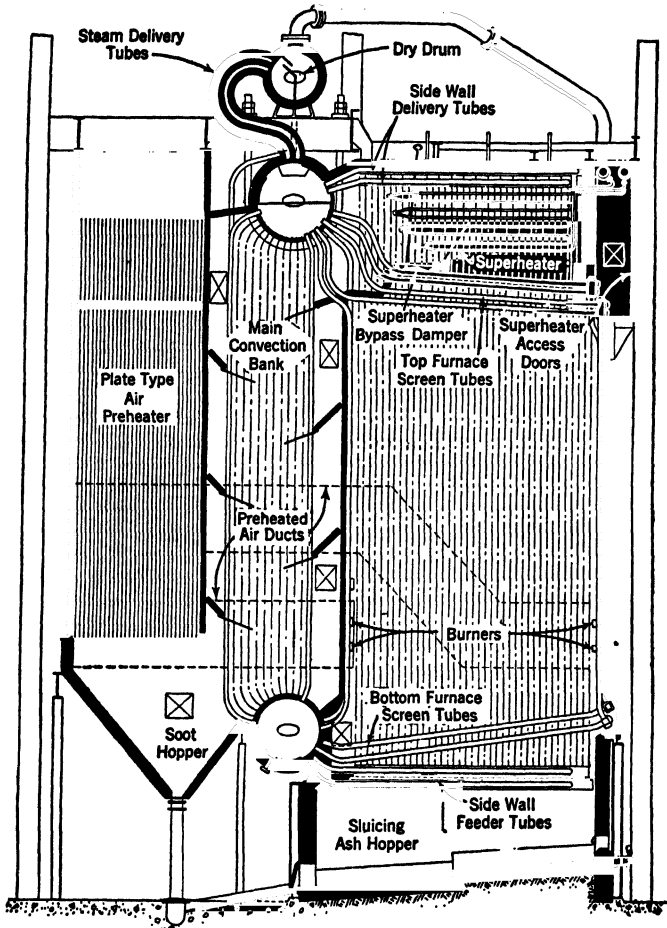



FIG. 40.—Combustion steam generator.

The water circulation is upward on all sides of the furnace and downward in the tubes of the convection section.

The *combustion steam generator* is noted for its rapid firing and high evaporating ability.

Figure 41 illustrates the *integral-furnace boiler unit* manufactured by the Babcock and Wilcox Company. This is shown with a part of the front wall and right side wall taken away, and looking toward

the bare-tube rear wall. Cut-away sections show the construction of the stud-tube side walls, the tubes of which enter the upper drum and are connected at the bottom either directly or indirectly through a bottom header to the lower drum. The water-cooled floor construction consists of water tubes covered by Bailey smooth metal blocks and supported by beams to hold the slag load.

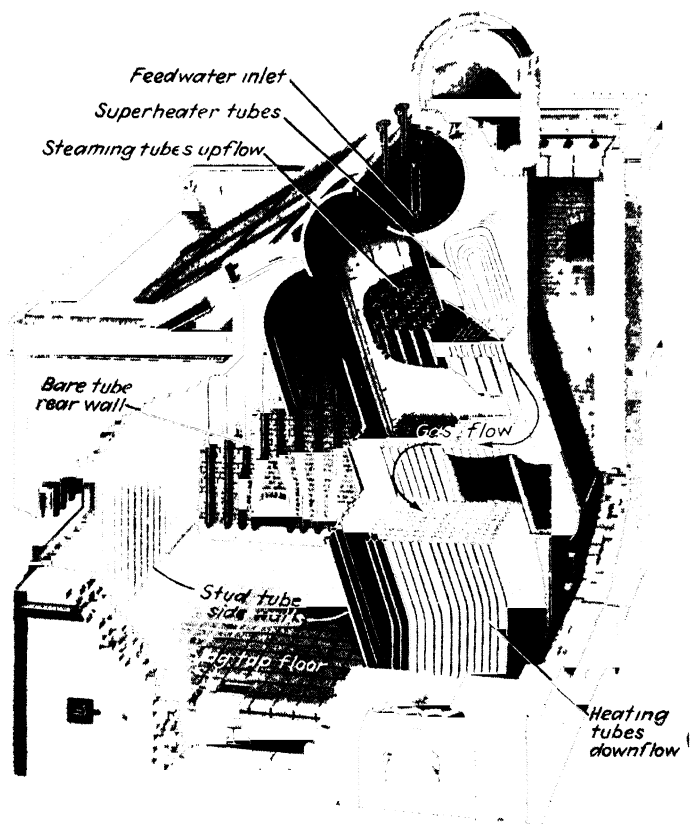



Fig. 41.—Integral furnace boiler unit. (*Babcock and Wilcox Company.*)

Gas passage is horizontal, the powdered-coal burners, not shown in the front wall which has been removed, directing the stream of burning coal downward toward the floor. The burning fuel and gases pass between the side walls toward the rear wall, where they turn and enter the steaming convection tubes. The stud-tube right side wall does not extend to the rear wall. The steaming tubes are made larger in diameter to allow for the increased circulation. The gases flowing

horizontally, pass across the steaming tubes, across the superheater loops, then around the end of the baffle returning toward the furnace. The gas flow is then through two passes of water-heating tubes, leaving the setting through an opening in the right side wall, or in the roof.

The boiler shown is designed for a pressure of 475 lb. and has a heating surface of 5,780 sq. ft. A tubular-type air preheater, not shown, has a heating surface of 5,780 sq. ft., composed of 2½-in. elements, 30 rows wide and 14 rows deep.

The water circulation is different from the usual bent-tube boiler. Water flows upward in the steaming tubes and downward in the water-heating tubes and longitudinally in the two drums.

Typical data and results with this type of boiler using West Virginia coal, 14,126 B.t.u. per pound as fired, are given in the following table:

TABLE 4-2.—PERFORMANCE DATA, BABCOCK AND WILCOX INTEGRAL-FURNACE BOILER UNIT (Fig. 41)

Steam, actual, 1,000 lb. per hour	50.0
Liberation, kB per cubic foot per hour	31.5
Solid fuel, coal, 1,000 lb. per hour	4.93
CO_2 leaving boiler, per cent	14.75
Steam pressure at drum, lb. per square inches	433
Second air pressure in burner casing, inches of water	4.0
Steam pressure at superheater outlet, lb. per square inch	425
Pressure drop drum to superheater outlet, lb. per square inch	8.0
Steam temperature, °F	705
Temperature of flue gas leaving boiler, °F	650
Temperature of flue gas leaving air heater, °F	406
Temperature of water entering boiler, °F	210
Temperature of air entering air heater, °F	80
Temperature of air leaving air heater, °F	423
Draft loss through boiler and superheater, inches of water	0.95
Draft loss through air heater, inches of water	1.45
Flue gas, 1,000 lb. per hour	69.2
Air through air heater, 1,000 lb. per hour	51.1
Excess air leaving boiler, per cent	25.0
Dry gas-heat loss, per cent	7.1
Hydrogen and moisture in fuel loss, per cent	4.0
Moisture in air loss, per cent	0.2
Unburned carbon loss, per cent	1.0
Radiation loss, per cent	0.8
Unaccounted for losses, per cent	1.5
Total losses, per cent	14.6
Efficiency of unit, per cent	85.4

Large boiler units indicate trends in design. Present design is largely influenced by three factors, high pressure, high temperature, and high capacity.

Drum construction for high-pressure boilers has changed from forged boiler drums to electric fusion-welded drums. The Boiler Code Committee in 1934 permitted drum calculations based on 70,000 lb. per square inch for steel plate of moderate thickness and 65,000 lb. per square inch for the heaviest plate Also formed drum heads may be welded to the ends of the drums The drum thickness is limited

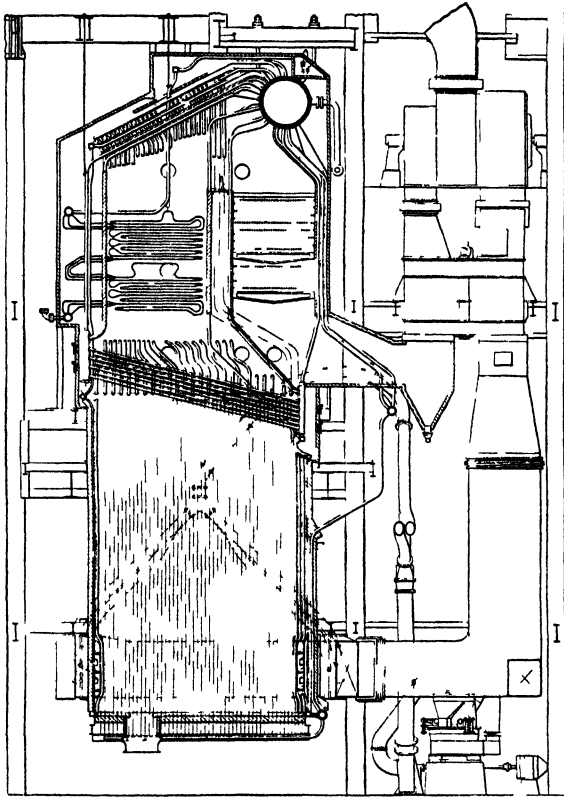



FIG 42 *a* —One of two 500,000-lb per hour, 1400-lb pressure 900°F steam temperature Combustion Engineering Company sectional-header boilers installed at Waterside Station of New York Edison Company

because plate must be accurately formed by cold bending. The drums for the Waterside Station (Fig 42 *a*), New York Edison Company, will be 60-in internal diameter, and $4\frac{3}{4}$ in thick The 54-in. drums for the 1,425-lb -pressure boiler of the Appalachian Electric Power Company to be installed at Logan, W. Va. (Fig. 42 *b*), will have a shell thickness of $3\frac{29}{32}$ in. This Duplex Boiler at the Logan Station has a generating capacity of one million pounds of steam per hour. Pulverized coal is burned in a dry-bottom furnace.

In the Waterside Station, the steam-generating unit produces 500,000 lb. of steam per hour at 1,400 lb. pressure, and 900°F. steam. The superheater is a major factor in the design, the service imposed upon the tubes being infinitely more severe than on the boiler.

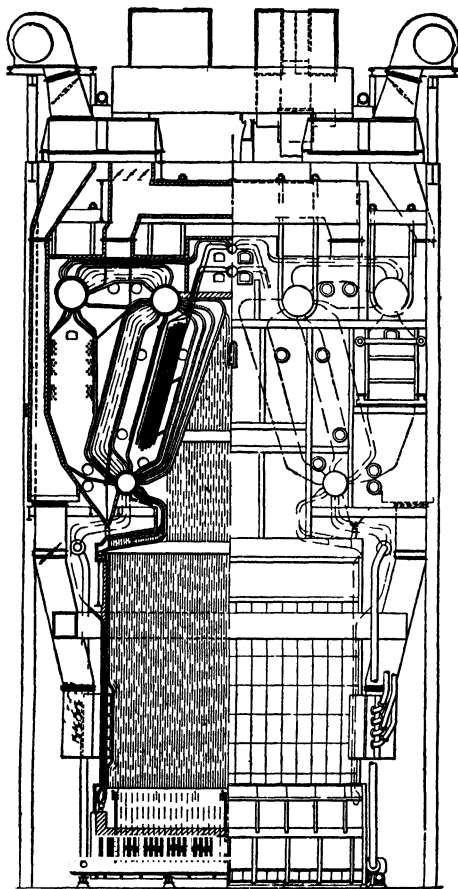



FIG 42 *b* —A 1,000,000-lb per hour, 1425-lb pressure, 925°F double-set bent-tube Combustion Engineering boiler installed at Logan, W Va , Station of the Appalachian Electric Power Company. Pulverized coal is burned in a dry-bottom furnace

In the Waterside design, the main part of the boiler is composed of the water walls surrounding the furnace. The convection portion of the boiler heating surface is little more than a screen. The gas temperature drop through the boiler tubes is about 200°F. The superheater follows the counterflow principle, steam flowing downward and gas upward. Hence the gas temperature drop through the super-

heater is around 1,000°F. The economizer and air preheater together account for a further drop of about 400°F. in the gas temperature.[1]

Figure 43 shows the portion of the heat absorption in 1 lb. of steam, assigned to the different equipment. It should be noted that of the absorption assigned to the boiler, 50 to 80 per cent is absorbed by the water walls.

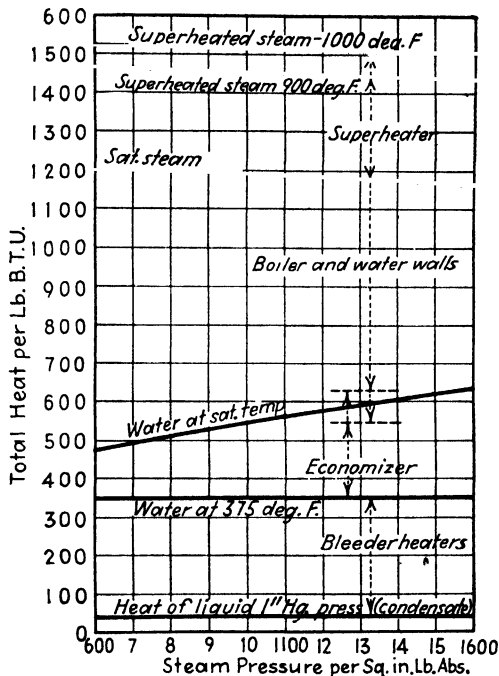



FIG. 43.—Allocation of heat absorption to various classes of high-pressure, heat absorbing equipment for pressures of 600 to 1600 lb. per square inch. (*Courtesy Combustion Engineering Company.*)

The complete unit includes the regenerative air preheater, unit pulverized-coal mill, and tangentially-corner-fired burners.

Figure 44 illustrates a sectional view of the 300,000-lb.-of-steam-per-hour, 1400-lb.-pressure, 760°F. steam-generating unit at the Firestone Tire and Rubber Company at Akron, O. The boiler is of the cross-drum, sectional-header type with a welded main drum, 54 in. diameter, 23 ft. long, 4 in. thick. There is an additional lower drum, increasing reserve water storage from 1.34 to 2.7 min. at full-load operation.

[1] Rosencrants, F. H., "Trends in design of large boiler units," Paper presented at Midwest Power Engineering Conference, Chicago, Ill. April 23, 1936.

The furnace is entirely surrounded by water walls backed with 2½ to 3 in. of insulating brick, then 3 in. of rock wool, inside the 10-gage steel casing. The top headers for the side walls are supported by steel beams, and the lower headers are hung on the water-wall tubes, permitting free expansion downward The furnace bottom is of

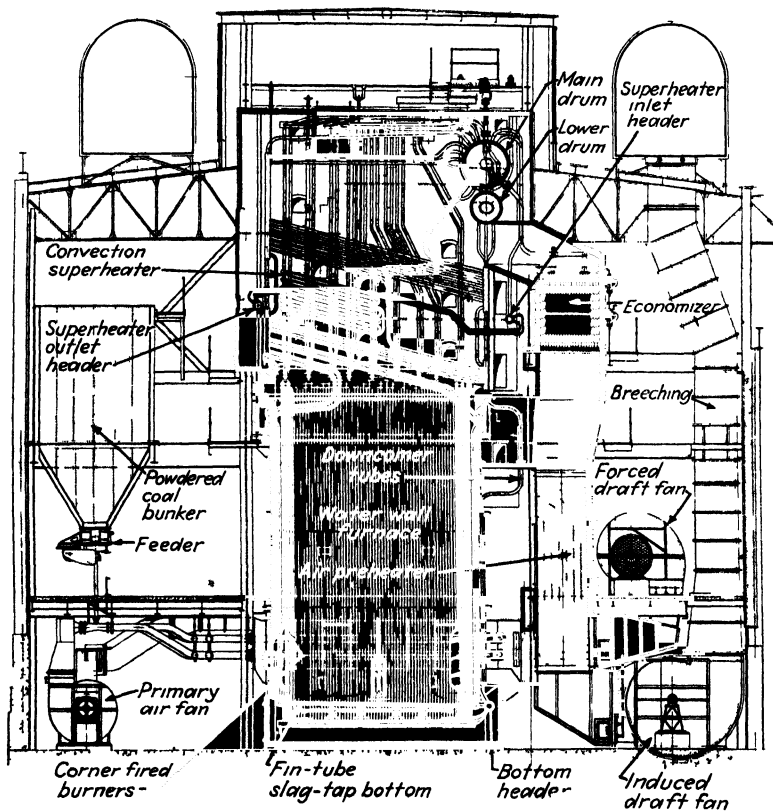



FIG 44—Section through 300,000-lb per hr, 1,400-lb pressure, CE sectional header boiler at Firestone Tire and Rubber Company, Akron Ohio This unit is fired by pulverized coal and has a slagging type of furnace (*Combustion Engineering Company, Inc*)

the fin-tube construction mounted on beams strong enough to carry the slag load. The furnace surface is connected into the boiler circulation, water from the rear boiler headers entering a separate header at the bottom of the furnace, which supplies the water tubes of the four walls and the bottom

The powdered coal is supplied from a central preparation plant Duplex feeders deliver coal from a 90-ton pulverized-coal bunker to

corner burners in each corner and near the bottom of the furnace. Both primary and secondary air are at a temperature of 444°F.

The economizer is of integral-loop construction with horizontal tubes. Gas flow is downward, and water flow upward. Water enters the economizer at 380°F. and leaves at 472°F., entering the main drum at two points, and is distributed along the full drum length to flow over steam-washing screens. Gas is cooled from 1020 to 680°F.

The air preheater, built of 12-gage copperoid vertical steel plates, welded together, is arranged for counterflow of gas and air. Gas temperature is lowered from 680 to 410°F., while the air is heated from 100 to 440°F.

This boiler supplies steam at 1250 lb. and 705°F. to a non-condensing turbine. The exhaust from the turbine at 235 lb. and 21° of superheat enters five evaporators operating in parallel. When the evaporators are supplied with 269,000 lb. per hour of dry steam at 235 lb. pressure, they produce 260,900 lb. per hour of steam at 180 lb. gage. The condensates from the evaporators return to the high-pressure circuit. Hence, the necessity for make-up water is eliminated, and no part of the high-pressure feedwater receives chemical treatment.

The following table gives the principal equipment of the steam-generating unit:

TABLE 4-3.—STEAM-GENERATING EQUIPMENT—FIRESTONE TIRE AND RUBBER COMPANY. 1,400-LB. BOILER (Fig. 44)

Boiler, Combustion Engineering Company: cross-drum: Sectional-header type, heating surface, sq. ft.: boiler, 11,793; front and rear walls, 1,697 each; side walls, 3,664 total; bottom, 526

Steam drum: 54 in. diameter, 23 ft. long, 4 in. thick; auxiliary drum, 36 in. diameter, 23 ft. long, 2 11/16 in. thick; furnace volume, 14,500 cu. ft.

Heat release at full load, 24,100 B.t.u. per hour per cubic foot.

Superheater, Superheater Company: Single-pass, multiple-loop, interdeck, 1 7/8 in. outside diameter seamless tubes, heating surface, 400 sq. ft.

Economizer, Combustion Engineering Company: Integral-loop, 2 in. outside diameter fin tubes, heating surface, 8,110 sq. ft.

Air heater, Combustion Engineering Company: Vertical plate, plate area, 23,100 sq. ft.

Pulverized-coal Burners, Combustion Engineering Company: Four duplex, corner-fired, tangential-type; pulverized-coal feeders driven by adjustable-speed motors.

Fans (drive, Westinghouse constant-speed 2,300-volt motors):

Primary air, Buffalo Forge Company: 21,000 c.f.m. at 440°F., 15 in. water, 1160 r.p.m., 100 hp.

Forced draft, Buffalo Forge Company: 103,000 c.f m., 100°F., 10 in. water, 1160 r.p.m., 250 hp.

Induced draft, Buffalo Forge Company: 183,000 c.f.m. 410°F., 11.5 in. water, 865 r.p.m., 500 hp.

63. Water Walls.—The tendency toward high temperatures which exceed the limits of refractory furnaces has brought a demand for water-cooled furnace walls. Water walls absorb mainly the surplus radiant heat that is, to a great extent, wasted in the ordinary refractory furnace, which results in improved boiler efficiency. Though not considered as a part of the boiler heating surface, water walls form a part of the boiler vessel.

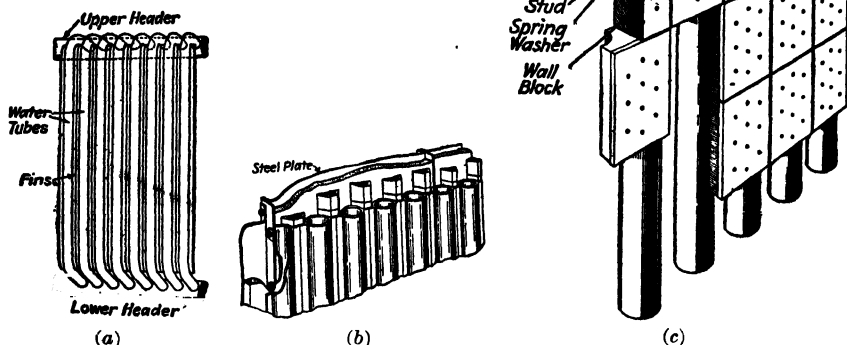



FIG. 45.—Water walls. (a) C-E Fin wall. (b) C-E Fin wall construction. (c) Bailey wall.

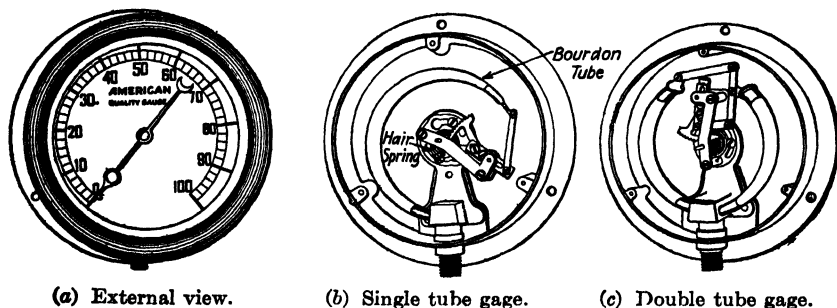
They consist of a series of parallel tube elements which extend, partly, within the furnace wall. The tube ends connect to upper and lower headers which usually connect with the steam drum and lower water drum, respectively, of the boiler. Figure 45 shows

(a) External view. (b) Single tube gage. (c) Double tube gage.

FIG. 46.—Bourdon tube pressure gage, showing internal features.

two types of widely used water walls. The heating surface of the water walls illustrated is the total surface of the tubes and fins exposed to the gases in the one case and, in the other, the exposed surface of the blocks.

64. Boiler Accessories.—Certain devices which, directly or indirectly, enter into the operation of steam boilers are called boiler accessories. Those which are direct-connected to the boiler and are really a part of it are a steam-pressure gage, water column, safety valve, blow-off valve, feedwater regulator and fusible plug. Incident to the boiler maintenance are soot blowers and tube cleaners.

Fig. 47.—Reliance water column, with high- and low-level alarm.

65. Steam-pressure Gage.—The steam-pressure gage most commonly used is of the Bourbon tube type, shown in Fig. 46. The main feature of this gage is the hollow bronze tube which has an oval cross-section, inside of which is exerted the boiler pressure. Figure 46 *b* shows the internal features of a gage having but one tube. This tube is bent in the arc of a circle and is connected at the lower end with the threaded steam connection below. The outer end is sealed off, and through a system of links it connects with the indicating pointer. The pressure inside tends to give the tube a circular cross-section, thus causing it to straighten somewhat and to actuate the pointer which indicates the pressure above atmospheric, in pounds per square inch, on the calibrated dial. For high pressures (above 500 lb. per square inch) two tubes of heavy material are usually used (Fig. 46 *c*). Using two tubes increases the linkage movement. As a protection against high temperatures due to the steam, a siphon tube which traps and holds condensed water is placed just below a gage.

66. Water Column.—To determine the water level in a boiler while under pressure, and in operation, a water column with water-gage glass is used. The smallest boilers use only a gage glass attached to the boiler shell. Figure 47 shows a type of water column commonly used on large boilers. It is placed, with respect to the boiler, so the desired average level of water will be indicated at about the middle of the water glass which should be visible at all times. The top and

bottom of the column are connected to the steam and water space, respectively, of the boiler, as shown in some of the Figs. 14 to 37. Both the column and try glass have blow-off connections for removing sediment. The try cocks on the side are used to check the water level at intervals. When located high and out of reach all the cocks of the column are operated by chains which hang to the boiler-room floor. The alarm feature is extra and consists of two floats, a lever cock, and a whistle. The lever arrangement is such that the cock is closed when the lower float is buoyed up by the water and when the upper float hangs by it's own weight. A fluctuation in water level above the

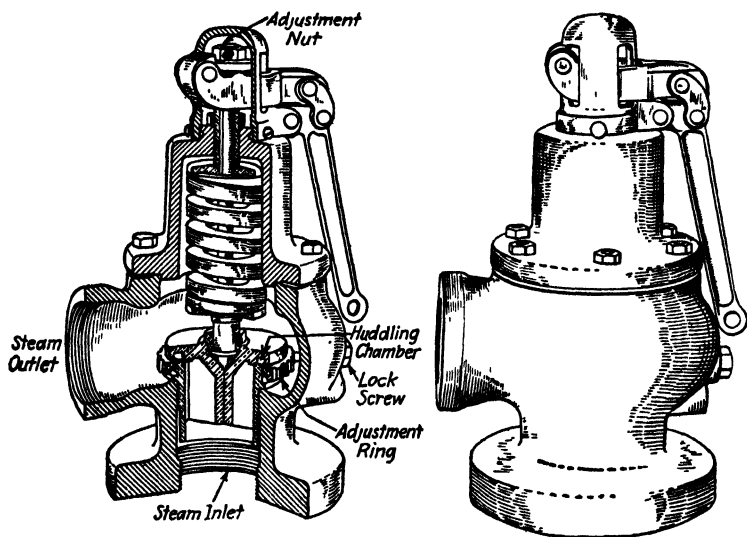



Fig. 48.—Consolidated pop safety valve.

upper or below the lower try cocks causes the upper float to rise or the lower float to fall, respectively. Either effect opens the cock and blows the whistle, indicating a faulty feedwater supply.

✓**67. Boiler Safety Valve.**—The standard boiler code requires that every boiler should have at least one safety valve attached to the main steam drum, with no intervening connections, and that it should be large enough to discharge the full steam capacity with not more than 6 per cent rise in pressure. A spring-loaded valve of the type shown in Fig. 48 is often specified. This valve consists essentially of a housing, valve disc, removable valve seat, and a spring which holds the disc in place. Popping is the result of two simultaneous events: first, a simmering of steam into the huddling chamber beneath the enlargement on the disc; and, second, an increased pressure in the

huddling chamber. The sudden exposure of the larger disc area to pressure causes the disc to pop open, allowing steam to escape. The spring may be adjusted for the desired pop pressure, and the adjusting

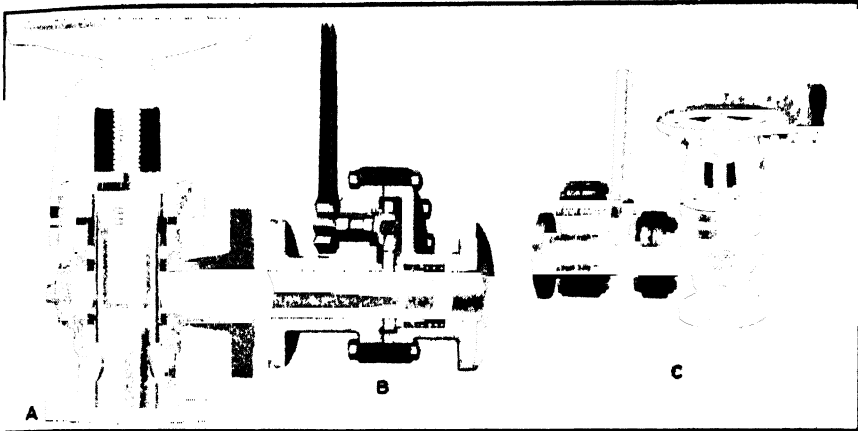



FIG. 49—Yarway blow-off valve in tandem with disc valve.

ring may be regulated to give the desired blow down or pressure drop. The A.S.M.E. Boiler Code requires a blow down of not less than 2 per cent of the opening pressure for steam pressures up to 300 lb. per square inch.

✓**68. Blow-off Valve.**—Figure 49 shows a type of blow-off valve commonly used in power plants. A lever-operated, quick-opening disc valve is placed next to the boiler, and in tandem with it is a balanced, seatless angle valve. When blowing down, the disc valve is first opened. The discharge is then regulated by the angle valve. A good blow-off valve should open easily and close tightly and be capable of discharging, freely, water heavily laden with scale and sediment, without injury to the valve parts.

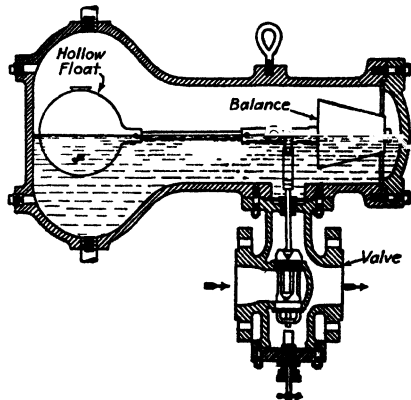



FIG. 50.—Stets boiler feedwater controller.

✓**69. Boiler Feedwater Controllers.**—The present tendency of boiler feed control is toward the use of automatic apparatus, which not only insures continuous feeding of supply water at the proper rate, but eliminates the constant attention necessary for hand control. Feedwater regulators in most common use are of the following types: float-

lever, thermoexpansion and thermopressure. The first two will be discussed here.

The float-lever type of feedwater regulator (Fig. 50) is mounted near the main steam drum with steam and water connections between the float chamber and the steam and water space, respectively, of the boiler. The mean water level in both the regulator and boiler corre-

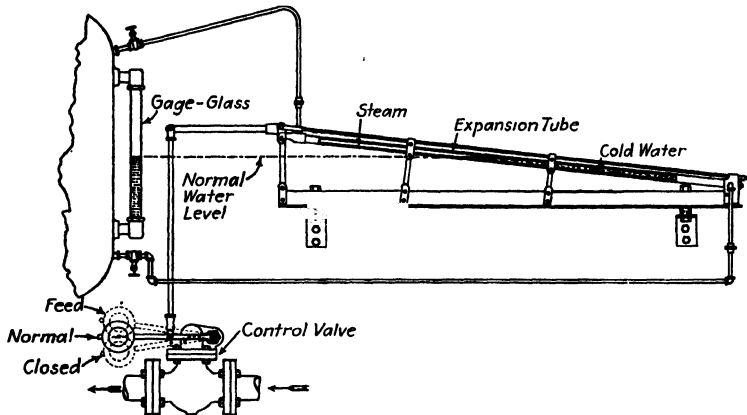



FIG. 51.—Copes feedwater regulator.

sponds. Fluctuations of the water level cause the float to rise or fall, and this action, through the connecting links, opens or closes the balanced valve in the feed line, thereby regulating the flow of water to the boiler. This type of regulator is quick to respond and is built for pressures up to 700 lb. per square inch.

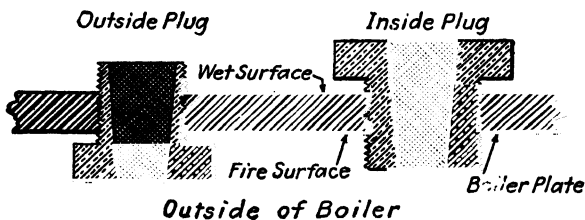



FIG. 52.—Fusible plugs.

The *Copes regulator* (Fig. 51) depends on the expansion and contraction of a heavy metallic inclined tube to operate the feed valve. The lower or stationary end of the tube is connected to the water space, and the upper or free end to the steam space of the boiler. Motion is transferred from the free end of the tube through the bell-crank lever, strut and valve lever to the balanced valve in the feed line. The tube and its supporting frame are mounted, as shown, so that with a mean level of water in the boiler, water will rise halfway

up the tube. Since there is no circulation this water will be at slightly above room temperature, and the upper part of the tube, which contains steam, will be at boiler temperature. A change in the water level exposes more or less of the tube to steam, causing a variation in its length and at the same time operation of the feed valve.

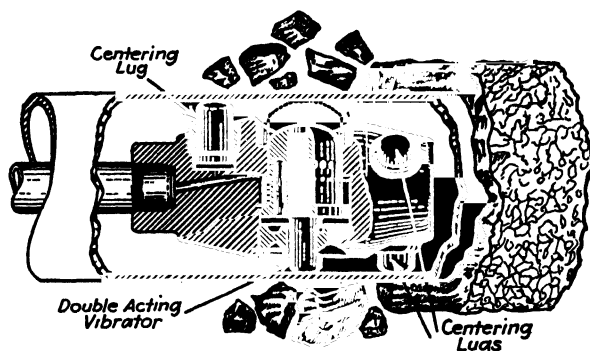



Fig. 53.—"Torpedo" fire-tube scale remover.

Feedwater regulators, when used in connection with excess pressure feed-pump governors, give most efficient service.

70. Fusible or Safety Plugs.—Fusible plugs (Fig. 52) are recommended for boilers operating below 225 lb. pressure. They are filled with a tin alloy which melts at between 400 and 500°F. and are usually put in the boiler shell so as to be exposed to steam when the water

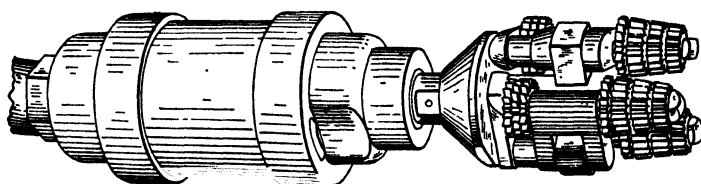



Fig. 54.—Lagonda water-tube scale remover.

level falls to a danger point. This exposure allows the hot gases to melt the alloy, which gives a hole for the escapement of steam. This signals the operator and deadens the combustion, avoiding possible damage.

71. Tube Cleaners.—Tube cleaners are highly important in the maintenance of boilers which use water containing scale-forming impurities. They are built in a variety of types and sizes and are either air, steam or water operated. The cleaner, with its hose, is thrust through the boiler tube, and by its knocking or cutting action removes the brittle scale formation. The knocker type of cleaner operates on steam or air and is used on fire-tube boilers only. Fig-

ure 53 shows one type of fire-tube cleaner. When the scale is on the inner surface of the tube, as in water-tube boilers, a cutter type of cleaner (Fig. 54) is used. The cutting elements are rotated by the attached motor and, by their rotative action, are thrown against the inner surface with sufficient force to cut the scale. It is often desirable to use a water motor with this type of cutter, as the discharged water may be used to wash away the scale as fast as it is removed. For use on fire-tubes, the cutters of the *Lagonda cleaner* are replaced by knocking or vibrating elements, which also rotate.

72. Soot Blowers.—Soot blowers also play an important part in the maintenance and efficient operation of a steam boiler. A coating

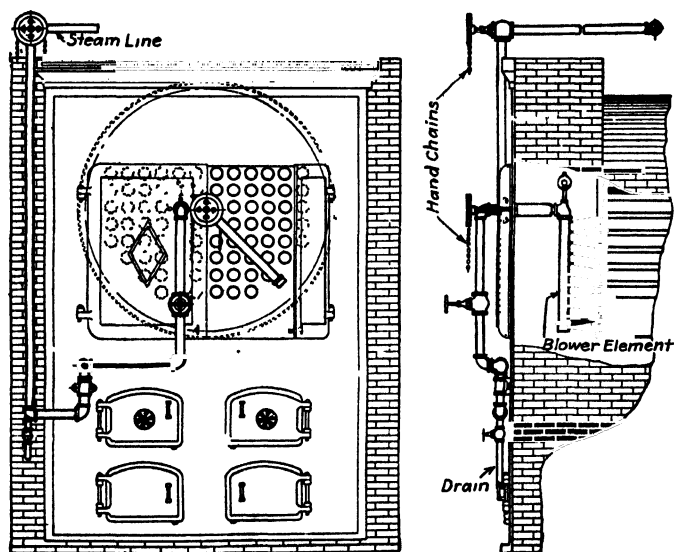



FIG. 55.—Diamond fire-tube soot blower.

of soot on the surface exposed to the gases is a good insulator and greatly restricts the flow of heat to the boiler. Fortunately, this coating is quite easily removed by brushes, hand-operated steam lances or permanently installed mechanical soot blowers. When lances or brushes are used, openings in the furnace walls provide access. It is often desirable, however, to have mechanical blowers and to use them at frequent intervals. Figure 55 shows one type of fire-tube blower installed for use on an H.R.T. boiler. As shown, steam is piped to the blower element which is a header for the jets along its length. By means of the attached sprocket and hand chain, the element may be swung to include all the tubes. The soot is blown through the tubes to the rear of the setting from where it can be

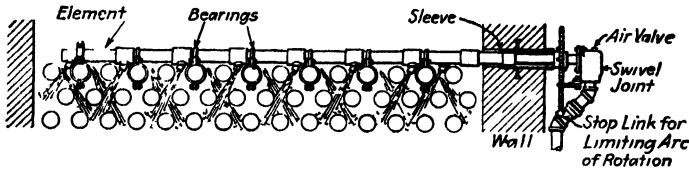
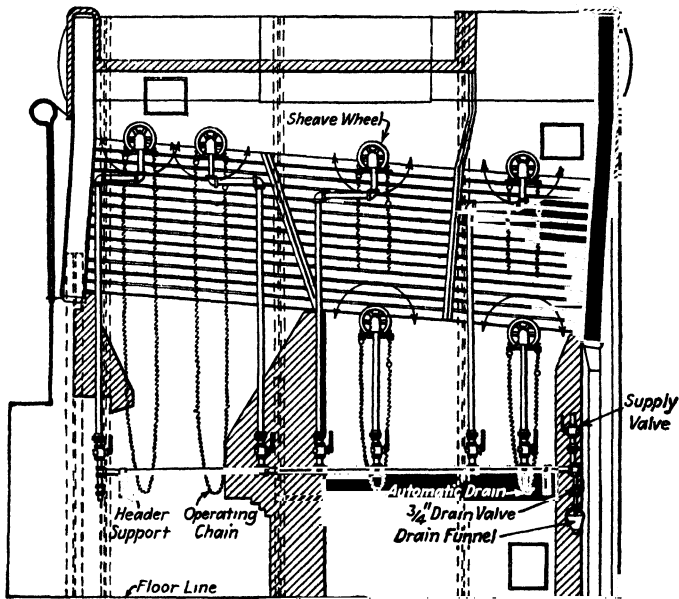
removed by shovels. In the case of water-tube boilers the soot must be blown from the outer surfaces. This may be accomplished by the steam blower shown in Fig. 56, or by a similar device. The elements extend across the tubes and are so located that all parts of the heating surface may be cleaned of any soot deposit. The attached sprocket and its hand chain afford a means of rotating the jets through a blow-

(a) Individual element.

(b) Showing installation of soot blowers.

FIG. 56.—Vulcan soot blowers.

ing arc of about 180 deg. Stops, fastened to the chain, determine the allowable arc. The soot, after being blown free, settles usually in the lower part of the setting, and from there it is removed through the clean-out doors.

73. Heat Transmission in Boilers.—The heat absorbed by a boiler depends on the amount and nature of the heating (or *heat-absorbing*) surface, the character of the plate, and the temperature difference between the hot gases and the exposed metal surface. The density

and velocity of the gases and the arrangement of the heating surface have a decided effect on these factors.

Boiler heating surface is the area of the dry plate and tube surface through which the hot gases and water have heat contact. Thus, the total dry surface below the mean water level that is in direct contact with the hot gases is that considered in determining the number of square feet of heating surface which a boiler may have. The remaining exposed and heated area is called *boiler superheating surface.* At present, *water-wall area* is not considered as a part of the boiler heating surface. It should be noted that the heating surface of fire tubes is on the inside, while, for water tubes, the outside area is considered.

FIG. 57.—Diagram showing heat transmission through boiler plate.

The dry plate or tube surface receives heat from the fire and furnace gases by

1. Radiation.
2. Convection.

The heat transfer through the boiler plate is by conduction, and the final emission of heat to the water is principally by convection.

The heat absorbed by radiation depends on the furnace temperature. In furnaces where combustion takes place rapidly, temperatures which cause fusion of a refractory lining are frequently reached, and this condition may prove both wasteful and destructive unless water walls or some other means of cooling is resorted to. On burning low-grade fuels having a relatively low combustion temperature, nearly all of the radiant heat is utilized in distilling and igniting the combustible gases.

Heat absorbed by convection results from the flowing of hot furnace gases over the heating surface. The specific heat, temperature and velocity of the gases are the principal factors in determining the amount of heat absorbed by this means.

The heat which flows through the boiler plate and surface coatings is finally carried away by the inner steam and water film to the main body of water. Figure 57 shows the character of the boiler plate. This figure also illustrates the generally accepted theory of heat transmission. The surface next to the gases is covered by a thin gas film and a coating of soot. A coating of scale and a steam film cover the opposite surface. It is sometimes considered that an arrangement of the heating surface so as to allow the inner surfaces

to be "scrubbed" by rising steam bubbles results in a thinner steam film, thereby increasing the evaporation over a given area.

The coefficient of heat flow is much lower for soot and scale than for steel. Considering this, it can be readily seen that coatings of soot and scale lower the heat flow through the boiler shell, thus lowering the boiler efficiency.

74. Builder's Rating.—When rating steam-power boilers, the manufacturers use the term "builder's" or "manufacturer's rating," a common term based wholly on the extent of the heating surface. Present practice considers 10 sq. ft. of *boiler heating surface* as a boiler horsepower. Thus, a boiler having 1,000 sq. ft. of *boiler heating surface* is rated as a 100-hp. boiler. This method of rating dates back to the early history of boilers. It places all boilers on the same basis, regardless of the fact that one may have its heating surface arranged to evaporate more water per square foot, under similar furnace conditions, than another. In the following, boiler horsepower based on evaporation is taken up, and it will be seen that a boiler may be operated at considerably above its rated horsepower. The combustion steam generator (Fig. 40, page 100) will evaporate approximately 50 lb. of water per square foot of heating surface, per hour, which corresponds to 1,500 per cent rating.

It should be noted that "horsepower" of a boiler is a misnomer and bears no direct relation to the term as used in connection with power machinery.

75. Equivalent Evaporation.—The *equivalent evaporation* is defined as the number of pounds per hour of water at 212°F. that could be evaporated to dry steam at 212°F., by the absorption of heat equal to that supplied per hour, to the feedwater, under actual conditions. Thus,

$$W_e = W \times \text{F.E.} \tag{63}$$

or

$$W_e = W\frac{(h_1 - h_{f2})}{970.2} \tag{64}$$

in which

W_e = equivalent evaporation, lb. per hour.

W = total weight (lb.) of feedwater actually evaporated per hour.

F.E. = factor of evaporation.

$$= \frac{h_1 - h_{f2}}{970.2} \tag{65}$$

h_1 = enthalpy of steam from steam-generating unit, B.t.u. per pound.

h_{f2} = enthalpy of water to steam-generating unit, B.t.u. per pound.

A comparison of the values of equivalent and actual evaporation gives a general idea of the boiler heat loss due to low-temperature feedwater.

76. Developed Boiler Horsepower.—The developed boiler horsepower is based on the unit of 1 boiler horsepower, which is defined as the evaporation of 34.5 lb. of water per hour, at 212°F. and 14.7 lb. per square inch absolute pressure. This is the equivalent of a heat absorption of 34.5 × 970.2 = 33,472 B.t.u. per hour.

The required data for determining the heat absorption in an actual case—such data as the weight and temperature of the water fed to the boiler unit, and the pressures and quality or superheat of the steam leaving—are found by test. The Steam Tables give the additional information necessary for calculation in a specific case.

The developed boiler horsepower is determined from the foregoing information and may be expressed by the following equations:

$$\text{Boiler horsepower} = \frac{W_e}{34.5} \tag{66}$$

$$= \frac{W \times \text{F.E.}}{34.5} \tag{67}$$

$$= \frac{W(h_1 - h_{f2})}{33{,}472} \tag{68}$$

$$= \frac{W(h_1 - h_{f2})}{34.5 \times 970.2} \tag{69}$$

in which the terms used are defined above.

Most boilers are operated at considerably above their nominal rating. In installations such as central and isolated power plants, the boiler ratings of 150 to 300 per cent are developed in continuous operation. The percentage of rating is obtained by dividing the horsepower actually developed by evaporation, by the rated or builder's horsepower. Thus

$$\text{Percentage of boiler rating} = \frac{\text{developed boiler horsepower} \times 100}{\text{builder's or nominal rating}} \tag{70}$$

With modern firing equipment the nominal rating is below that which gives the highest boiler efficiency. The point of most efficient operation is determined by test.

77. Performance.—The performance of any steam-generating unit is determined by the results of an operating test. The instruc-

tions for running a test, taking data, and calculating results are given in complete detail in the A.S.M.E. Boiler Test Code for 1936. The unit of heat absorption, used in calculating certain results, is taken as 1,000 B.t.u. and expressed by the symbol kB. From a complete performance test, the following items may be determined:

1. *Rate of Heat Absorption by Steam-generating Unit.*—This is determined by the following equation:

$$Q_T = \frac{W(h_2 - h_1)}{1{,}000} \tag{71}$$

in which

Q_T = overall rate of heat absorption by steam-generating unit, kB per hour.

W = weight of water evaporated, lb. per hour.

h_1 and h_2 = enthalpy of water entering the unit, and enthalpy of steam leaving unit, respectively, B.t.u. per pound.

When the heat absorbed is determined for each part of the unit separately, Q_T is expressed as follows:

$$Q_T = \Sigma(Q_1 + Q_2 + Q_3 + \cdots)$$

in which

Q_1, Q_2, etc., = the heat absorbed by each part of the unit, kB per hour.

The values for Q_1, Q_2, etc., are determined as in Eq. (71), where the terms used represent data for the water or steam, as the case may be, for the particular part of the unit.

2. *Rate of Heat Absorption per Pound of Fuel.*—This is determined by dividing the value obtained for Q_T in Eq. (71) by W_f, when W_f equals the weight of fuel (as fired or dry) consumed by the unit per hour in pounds (if the fuel is gas, this item is on a cubic-foot basis at standard conditions).

3. *Rate of Heat Absorption per Square Foot of Heating Surface.*—To obtain this, divide the value obtained for Q_T in Eq. (71) by A_T, when A_T equals the total number of square feet of heating surface in the steam-generating unit, excluding the air heater.

The heating surface of the steam-generating unit includes that portion of the surface of the heat-transfer apparatus exposed on one side to the combustion gases (or to the refractory being cooled) and on the other to the fluid being heated, measured on the side receiving heat. In relation to the separate parts of the unit, it is the sum of the areas of heating surface in each of the following: boiler,

water walls, water screens, water floor, superheater, economizer, and reheater.

✓4. *Efficiency of Steam-generating Unit.*—The efficiency is expressed by the following equation:

$$e = \frac{W(h_2 - h_1)}{W_f \times F} \times 100 \qquad (72)$$

in which

e = efficiency, per cent.

F = higher heat value of fuel, as fired (or dry, if W_f is dry), B.t.u. per pound.

Other symbols as in Eq. (71).

If an air heater is included in the unit, the heat absorbed therein is charged against the unit. In solving for the efficiency to apply in this case the following equation may be used:

$$e' = \frac{W(h_2 - h_1)}{W_f \times F + W_a \times 0.24(t_4 - 70)} \times 100 \qquad (73)$$

in which

e' = efficiency, per cent.

W_a = weight of air supplied per hour, lb.

t_4 = temperature of air supplied for combustion, °F.

Other symbols as in above equations.

5. *Rate of Combustion.*—This is expressed as pounds of fuel (dry or as fired) per hour; per square foot of projected grate area; per square foot of retort, or per retort; per burner (burner used with pulverized, liquid and gaseous fuels); or per cubic feet of furnace volume. Any of these are obtained by a simple calculation.

6. *Combustion space (cubic feet) per pound of fuel per hour.*

7. *Heat liberated (B.t.u.) per cubic foot of combustion space of furnace volume.*

The rate of combustion is one of the limiting factors in the developed boiler horsepower and depends on the kind and size of fuel, the grate, the air supplied, and the overall efficiency. An actual test is the best way to determine the most economical combustion rate for any specific case. With natural draft, 5 to 30 lb. of granular coal per square foot of grate area per hour can be burned with success. In mechanical-draft systems the rate is greatly increased. A maximum rate of 225 has been obtained in locomotives for short intervals.

With the proper proportion of fuel and air, hand-fired boilers may be operated continuously at efficiencies of 45 to 65 per cent. When stokers are used, 60 to 88 per cent is obtainable. Oil fuel gives values

slightly greater than the average for crushed coal. The highest efficiencies, 80 to 92 per cent, are generally reached in new installations using pulverized coal or underfeed stokers. The maximum efficiency, in any case, may be obtained in operation at considerably above

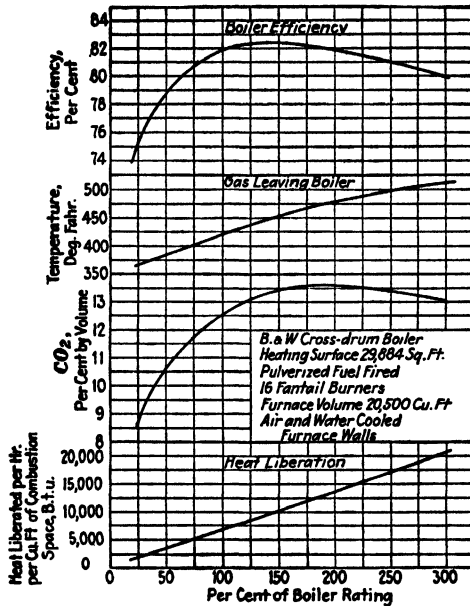



FIG. 58.—Typical boiler-performance curves.

nominal rating. The curves shown in Fig. 58 are typical of boiler performance in the average power plant.

Example 4-1.—The following data were taken in a test of a boiler and superheater unit using a six-retort underfeed stoker.

Boiler heating surface, sq. ft.	6,250
Grate surface, sq. ft.	120.29
Water fed to boiler per hour, lb	29,296
Temperature feedwater, °F	180
Coal, as fired per hour, lb	3,459
Higher heat value of coal, B.t.u. per pound (as fired)	11,968
Steam pressure, lb. per square inch absolute	191
Steam temperature, °F	504

It is required to find (*a*) the heat absorption (*kB* per hour), (*b*) equivalent evaporation per hour, (*c*) builder's rating, (*d*) developed boiler horsepower, (*e*) percentage of rating developed, (*f*) overall efficiency, and (*g*) combustion rate (as-fired basis).

Solution.

$$(a)\ \text{Heat absorption} = \frac{29{,}296(1270.94 - 147.87)}{1{,}000} = 32{,}900\ kB \text{ per hour.}$$

(*b*) $W_e = 29{,}296 \times \left(\frac{1270.94 - 147.87}{970.2}\right) = 33{,}840$ lb. per hour.

(*c*) Builder's rating $= \frac{6{,}250}{10} = 625$ boiler hp.

(*d*) Developed boiler horsepower $= \frac{W_e}{34.5} = \frac{33{,}840}{34.5} = 981.$

(*e*) Percentage of rating developed $= {}^{981}\!/_{625} \times 100 = 156.8.$

(*f*) Overall efficiency $= \frac{W(h - h_{f2})}{W_f \times F} = \frac{29{,}296(1{,}270.94 - 147.87)}{3{,}459 \times 11{,}968} \times 100$

$= 79.62$ per cent.

(*g*) Combustion rate $= \frac{3{,}459}{120.29} = 28.7$ lb. per square foot of grate per hour

or

$= \frac{3{,}459}{6} = 576.5$ lb. per retort per hour.

78. Heat Balance for a Steam-generating Unit.—The loss or unutilized portion of the heat resulting from the combustion of fuel in the furnace of the steam-generating unit is due to various causes which, to some extent, cannot be avoided. The major causes may be defined and classified and the consequent heat loss calculated. A tabulation or chart of the losses, together with the heat absorbed by the unit, constitutes the *heat balance* for a steam-generating unit. The items usually considered are listed as follows:

1. Heat absorbed by the steam-generating unit.
2. Heat loss due to moisture in coal.
3. Heat loss due to moisture resulting from combustion of hydrogen.
4. Heat loss due to moisture in the air.
5. Heat loss due to dry chimney gases.
6. Heat loss due to unburned combustible gases.
7. Heat loss due to unconsumed combustible in the refuse.
8. Heat loss due to unconsumed hydrogen and hydrocarbon compounds in chimney gases. Losses due to radiation and the losses from unaccounted-for causes are generally included in this item also.

Computation of the above items is usually made so that the results are expressed as B.t.u. per pound of solid or liquid fuel, or per cubic foot of gas, and as per cent of the heat supplied. Thus, the total of all the items should be the higher heat value of the fuel and 100 per cent, respectively.

A heat balance of this type may be made with the fuel analysis given on the as-fired.or dry basis. In computing, the basis used should be followed consistently throughout.

The main purpose of the heat balance is to show the distribution of the heat supplied in each pound of fuel. It also offers a means for study of each item, individually.

1. *Heat Absorbed by the Steam-generating Unit.*—The heat thus absorbed may be divided to show the amount of heat absorbed by each part of the equipment, provided sufficient data are available. Calculation of this item may be made using the following expression:

$$\frac{W(h_2 - h_1)}{W_f} \tag{74}$$

in which the heat absorbed is given in B.t.u. per pound of fuel, and

W = amount of water evaporated, lb. per hour.
W_f = fuel burned, lb. per hour.
h_1 = enthalpy of water entering unit, B.t.u. per pound.
h_2 = enthalpy of steam leaving unit, B.t.u. per pound.

2. *Heat Loss Due to Evaporation and Superheating of Moisture in the Fuel.*—This loss is comparatively low, except in a case where fuels of high moisture content are used, as with wood and baggasse. The following expressions may be used in computing:

$$\left.\begin{aligned} &\frac{M}{100}(1066 + 0.5t_g - t_a) \qquad \text{when } t_g > 575°\text{F.} \\ &\frac{M}{100}(1089 + 0.46t_g - t_a) \qquad \text{when } t_g < 575°\text{F.} \end{aligned}\right\} \tag{75}$$

in which the heat loss is found in B.t.u. per pound of fuel, and

M = moisture, from proximate analysis, per cent by weight.
t_g = temperature of gases leaving unit, °F.
t_a = temperature of air supplied for combustion, °F.

3. *Heat Loss Due to Evaporating and Superheating of Moisture Formed by the Combustion of Hydrogen.*—The hydrogen involved in this loss should not include that inherent in the moisture of the fuel. If the hydrogen as given in the ultimate analysis does include that in the moisture, as is frequently the case, the latter hydrogen, which is $M \div 9$, must deducted to obtain the value to be used.

This loss usually ranges from 2.5 per cent with the best coals to 14 per cent with coke-oven gas. It may be expressed by

$$\left.\begin{aligned} &\frac{9\text{H}}{100}(1{,}066 + 0.5t_g - t_a) \qquad \text{when } t_g > 575°\text{F.} \\ &\frac{9\text{H}}{100}(1{,}089 + 0.46t_g - t_a) \qquad \text{when } t_g < 575°\text{F.} \end{aligned}\right\} \tag{76}$$

in which H = per cent of hydrogen in fuel (from ultimate analysis). Other symbols as in item 2.

4. *Heat Loss Due to Superheating the Moisture in the Air Supplied for Combustion.*—Under ordinary conditions, the value of this loss is small and is frequently neglected. To solve for it, use the expression:

$$0.47w_v(t_g - t_a) \tag{77}$$

in which t_g and t_a are as in item 2, and

$$w_v = rdvw_a$$

where

w_v = weight of water vapor in air supplied for combustion, lb. per pound of fuel.

r = relative humidity (decimal) of air supplied, at temperature t_a.

d = weight of water vapor to saturate 1 cu. ft. of dry air at temperature t_a, lb. (Table 4-4, page 127).

v = volume of 1 lb. dry air at temperature t_a (Table 4-4).

w_a = weight of dry air supplied per pound of fuel (see Art. 49, page 72).

5. *Loss Due to Heat Carried Away in the Dry Chimney Gases.*—This loss is generally greater than any other and can be controlled, within limits, by lowering w_g and t_g and raising t_a as they are defined in the following expression which is used in determining the amount of heat lost in this way:

$$0.24w_g(t_g - t_a) \tag{78}$$

in which

w_g = weight of dry chimney gas per pound of fuel, lb.

and t_g and t_a are as in item 2.

6. *Heat Loss Due to Incomplete Combustion.*—This loss results from a failure to completely burn the carbon of the fuel, that is, to CO_2 and is due to insufficient air or improper mixing the air and combustible gases in the furnace. Under extremely poor furnace conditions the per cent of CO in the flue gas may be high and the resulting heat loss correspondingly high. With favorable conditions it is very low—very often zero. The expression for it is:

$$C\left(\frac{10{,}160CO}{CO_2 + CO}\right) \tag{79}$$

in which

C = weight of carbon burned per pound of fuel, lb. (see Art. 48, page 70).

CO = per cent, by volume, in dry gas leaving unit.

CO_2 = per cent, by volume, in dry gas leaving unit.

7. *Heat Loss Due to Unconsumed Combustible in the Ash and Refuse.* Poorly designed grates and improper firing methods result in considerable waste of combustible. This loss may be detected by the presence of carbon in the ash and refuse and is determined by the expression below.

$$14{,}600\left(\frac{W_a}{W_f} - \frac{A}{100}\right) \qquad (80)$$

in which

W_a = weight of ash and refuse per hour, lb.
W_f = weight of fuel per hour, lb.
A = percentage of ash, from ultimate analysis.

8. *Heat Loss Due to Unconsumed Hydrogen and Hydrocarbon Compounds in the Chimney Gases. Also Losses Due to Radiation and from Unaccounted-for Causes.*—Unconsumed hydrocarbon compounds are indicated by the presence of smoke in the chimney gases and result from a decomposition of the unstable hydrocarbon constituents of the fuel, due, principally, to an insufficient air supply at the point of distillation. The resulting heat loss can be determined accurately from a complete chemical analysis of the chimney gases or estimated from an available chart.

It is impossible to obtain data for the calculation of all the losses which occur in the operation of a steam-generating unit. Many of them are small and result from unknown causes. For these reasons the losses under this heading are included in one item and determined by difference, using the expression

$$F - (\text{sum of items 1 to 7, inclusive})$$

in which

F = higher heat value of the fuel, B.t.u. per pound.

In a carefully performed heat-balance test the amount of this loss should be less than 2 per cent. It is possible, however, that this item may turn out to be negative, which indicates the difficulty of obtaining truly representative values of data, such, for instance, as gas temperatures or gas analysis.

Example 4-2.—Calculation of the items for a heat balance, in Example 4-1 (page 121), is to be made, using the following additional data taken from an actual test.

Ultimate analysis of coal as fired, percentage by weight:

Carbon	66.85	Sulphur	3.06
Oxygen	6.13	Free moisture	6.00
Hydrogen	4.28	Ash	12.50
Nitrogen	1.18		
		Total	100.00

Flue-gas analysis, percentage by volume:

CO_2	12.80	CO	0.10
O_2	6.20	N_2(by difference)	80.90

Miscellaneous data:

Temperature of coal and air entering furnace, °F	75.2
Temperature of flue gases, °F	496.6
Relative humidity of draft air, per cent	55.0
Dry ash and refuse per hour, lb	520.0
Combustible in dry ash, per cent	16.7

Solution.—Carbon burned per pound of coal is (3,459 × 0.6685 − 520 × 0.167) ÷ 3,459 = 0.644 lb.
Weight of dry flue gas per pound of coal is [from Eq. (59), page 72]

$$w_g = 0.644 \times \frac{4 \times 12.8 + 6.2 + 700}{3(12.8 + 0.1)} = 12.59 \text{ lb.}$$

Weight of dry air supplied per pound of coal is [from Eq. (61), page 72]

$$w_a = 0.644 \times \frac{3.03 \times 80.90}{(12.8 + 0.1)} = 12.21 \text{ lb.}$$

Weight of water vapor in air supplied, per pound fuel is

$$w_v = 0.55 \times 0.00136 \times 13.48 \times 12.21 = 0.1231 \text{ lb.}$$

(Values 0.00136 and 13.48 from Table 4-4).

HEAT LOSSES PER POUND OF COAL AS FIRED

Item	Name of item or loss	Calculation	B.t.u. per pound	Per cent
1	Heat absorbed by boiler	$\frac{29,296(1,270.94 - 147.87)}{3,459}$	9,530	79.63
2	Moisture in fuel	(6 ÷ 100)(1,089 - 75.2 + 0.46 × 496.6)	75	0.63
3	Combustion of hydrogen	9(4.28 ÷ 100)(1,089 − 75.2 + 0.46 × 496.6)	477	3.99
4	Moisture in draft air	0.47 × 0.1231(496.6 − 75.2)	25	0.21
5	Dry chimney gas	12.59 × 0.24(496.6 − 75.2)	1,270	10.60
6	Incomplete combustion	(0.65 × 10,160 × 0.001) ÷ (0.128 + 0.001)	51	0.43
7	Combustible in ash	$\left(\frac{520}{3,459} - \frac{12.50}{100}\right)14,600$	365	3.05
8	Radiation, etc	By difference	175	1.46
Total			11,968	100.00

TABLE 4-4.—DATA FOR MIXTURES OF AIR AND SATURATED WATER VAPOR[1]

Temperature, °F.	Weight of saturated vapor		Volume, cu. ft.	
	Lb. per cu. ft. of dry air	Lb. per lb. of dry air	Of 1 lb. of dry air	Of 1 lb. of dry air + vapor to saturate it
0	0.0000674	0.000781	11.58	11.59
6	0.0000909	0.001067	11.73	11.75
10	0.0001103	0.001309	11.83	11.86
16	0.000147	0.001764	11.99	12.02
20	0.000177	0.002144	12.09	12.13
26	0.000232	0.002854	12.24	12.30
30	0.000278	0.003444	12.34	12.41
35	0.000340	0.004268	12.47	12.55
40	0.000410	0.005202	12.59	12.70
45	0.000492	0.00632	12.72	12.85
50	0.000588	0.00764	12.84	13.00
55	0.000699	0.00920	12.97	13.16
60	0.000829	0.01105	13.10	13.33
65	0.000979	0.01323	13.22	13.50
70	0.001153	0.01578	13.35	13.69
75	0.001352	0.01877	13.48	13.88
80	0.001580	0.02226	13.60	14.09
85	0.001841	0.02634	13.73	14.31
90	0.002137	0.03109	13.86	14.55
95	0.002474	0.03662	13.98	14.80
100	0.002855	0.04305	14.11	15.08
105	0.003285	0.0505	14.24	15.39
110	0.003769	0.0593	14.36	15.73
115	0.004312	0.0694	14.49	16.10
120	0.004920	0.0813	14.62	16.52
125	0.005599	0.0953	14.75	16.99
130	0.006356	0.1114	14.88	17.53
135	0.007197	0.1305	15.00	18.13
140	0.008130	0.1532	15.13	18.84
145	0.00916	0.1800	15.26	19.64
150	0.01030	0.2122	15.39	20.60
155	0.01156	0.2511	15.52	21.73
160	0.01294	0.2987	15.64	23.09
165	0.01445	0.3577	15.77	24.75
170	0.01611	0.4324	15.90	26.84
175	0.01793	0.5290	16.03	29.51
180	0.01991	0.6577	16.16	33.04
185	0.02206	0.8359	16.28	37.89
190	0.02441	1.0985	16.41	45.00
200	0.02972	2.2953	16.67	77.24

[1] (Reprinted by permission from "Properties of Steam and Ammonia," by Goodenough, published by John Wiley & Sons, Inc.)

The heat losses given above are considered for continuous operating efficiency of the boiler, disregarding the economic losses, such as for the frequent blow down and stand-by or banking periods. There is a certain portion of the actual loss that is inevitable or inherent. This is called the *inherent loss* and is due to heating the flue gases, moisture in the fuel, moisture in draft air, and the moisture from the combustion of hydrogen from boiler room to steam or heating surface temperature. These items may be calculated from the usual data, and from the results a maximum theoretical boiler efficiency may be determined

Problems

1. A boiler generates 13,611 lb. of dry steam per hour, at 206-lb. gage pressure. The barometer reading is 28.9 in. of mercury, and the feedwater enters the boiler at 189°F. Calculate (*a*) the factor of evaporation, (*b*) equivalent evaporation, per hour, and (*c*) the developed boiler horsepower.

2. Under test a boiler generated 37,164 lb. of 98 per cent dry steam per hour; pressure 306-lb. gage; and feedwater 205°F.; barometer 29.1 in. of mercury. Calculate (*a*) the equivalent evaporation per hour, (*b*) the developed boiler horsepower.

3. Solve Probiem 2, considering the steam to be passed through a superheater and heated to 625°F., at constant pressure. Determine the horsepower developed by the boiler and superheater.

4. A marine water-tube boiler having 2,500 sq. ft. of heating surface generates 11,124 lb. of 99 per cent dry steam per hour, at 199-lb. gage pressure. The barometer reads 29.4 in. of mercury; feedwater temperature 66°F.; dry coal used per hour 1,150 lb.; heat value of dry coal 14,040 B.t.u. per pound. Calculate (*a*) the developed boiler horsepower, (*b*) percentage of rating developed, (*c*) overall boiler efficiency and (*d*) the combustion rate, if the grate area is 78 sq. ft.

5. The following data were taken from the report of a test on a water-tube boiler with a superheater: Dry coal fired per hour 13,670 lb.; feedwater per hour 133,600 lb.; feedwater temperature 250°F.; steam pressure, 205 lb., gage; temperature of steam 650°F.; barometer 29.6 in. of mercury; heat value of fuel 14,125 B.t.u. per pound; boiler heating surface 15,000 sq. ft. Calculate (*a*) factor of evaporation, (*b*) equivalent evaporation per hour, (*c*) percentage of rating developed, (*d*) efficiency of boiler, furnace, grate and superheater, (*e*) cost per 1,000 lb. of steam actually evaporated if the coal costs $3.25 per ton and contains 8 per cent moisture.

6. Using the following boiler test data, tabulate the items of a boiler heat balance and show method of calculation for each item: Duration of test 48 hr.; coal fired 77,490 lb.; dry ash and refuse 6,375 lb.; feedwater 663,300 lb. *Pressures*, pounds gage: boiler 180; calorimeter 4.5. Barometer 29.6 in. of mercury; heat value of coal 13,950 B.t.u. per pound as fired. *Temperatures*, °F.: calorimeter 294; feedwater 205; stack 822; room 85. *Ultimate analysis* of coal fired, percentage: C 78; H_2 5.24; S 0.95; N_2 1.23; O_2 7.48; ash 7.10. *Proximate analysis* of coal fired, percentage: M 2.5; V.M. 21.4; F.C. 69.0; ash 7.10. *Ash analysis*, percentage: ash 86.3; combustible 13.7, *Flue-gas analysis*, percentage by volume: CO_2 11.45; O_2 7.91; CO 0.06. Humidity, percentage 60.

7. Calculate, from the data in Problem 6, (*a*) the equivalent evaporation, (*b*) the developed boiler horsepower and (*c*) overall boiler efficiency.

8. The following data were taken from a report of a test on a pulverized-fuel-fired water-tube boiler with superheater: Boiler heating surface 8,220 sq. ft.; furnace volume 3,700 cu. ft.; 6 vertical burners used. Steam pressure 190-lb. gage; barometer 29.4 in. of mercury; room air humidity 59 per cent. *Temperatures,* °F.: steam 399; feedwater 102; flue gases 731.7; room 85. *Flue-gas analysis,* percentage: CO_2 14.5; O_2 4.6; CO 0.1. *Ultimate analysis* of coal fired: S 1.44; H_2 5.38; C 74.27; N_2 1.89; O_2 10.46; ash 6.56. *Proximate analysis* of coal fired: M 3.89; V.M. 38.14; F.C. 51.41; ash 6.56; heat value 13,356 B.t.u. per lb. as fired. Fly ash 162.5 lb. per hour; coal per hour 2,480 lb.; feedwater per hour 20,300 lb. Calculate (*a*) pounds coal per cubic foot of furnace volume, (*b*) heat liberated per cubic foot of furnace volume and (*c*) complete heat balance, and show calculations.

9. A test on boiler No. 4 in the Purdue University power plant gave the following data: Boiler heating surface 5,006 sq. ft.; grate surface 103 sq. ft.; 5-retort underfeed stoker; furnace volume 2,110 cu. ft. *Pressures,* pound gage: superheater 192. Barometer 28.7 in. of mercury. *Temperatures,* °F.: superheater 512; combustion air 76; flue gas 540; feedwater 161; room air 74. *Flue-gas analysis,* percentage: CO_2 8.75; O_2 9.75; N_2 81.5. *Ultimate analysis,* percentage: C 64; H_2 5.41; O_2 18.17; N_2 1.41; S 1.31; ash 9.7. *Proximate analysis,* percentage: V.M. 32.4; F.C. 46.5; M 11.4; ash 9.7. Heat value of coal burned 11,030 B.t.u. per lb. as fired; moisture in air, per pound, 0.0116 lb.; coal fired per hour 2,521 lb.; dry ash and refuse per hour 292 lb.; feedwater per hour 17,800 lb.; combustible in dry ash and refuse 16.2 per cent. Calculate:

a. Combustion rate, pounds of coal per square foot of grate per hour.
b. Combustion rate per retort per hour.
c. Factor of evaporation.
d. Equivalent evaporation per hour.
e. Developed boiler horsepower.
f. Percentage of boiler rating.
g. Heat liberated per cubic foot of furnace volume.

10. Using the data in Problem 9, calculate the complete boiler heat balance and show the calculations for each item.

11. *Boiler Test Data:* Hourly Quantities—actual steam 402,700 lb.; coal, as fired 47,859 lb.; ash and refuse 3,562 lb.

Temperatures (°F.) and Pressures (pounds per square inch absolute)—steam leaving superheater, 660°F., 290 lb.; feedwater to boiler 262°F.; fuel and air to furnace 131°F.; gas leaving boiler 504°F.

Other Data—coal used, Cambria County, Pa.; flue-gas analysis CO_2 13.5, O_2 5.0, CO 0.8 per cent; combustible in ash and refuse 14 per cent; relative humidity 35 per cent (air to furnace).

Calculate the following items of heat balance, B.t.u. per pound of coal as fired, and as per cent of heating value of coal:

a. Heat absorbed by boiler and superheater.
b. Heat loss due to moisture in air.
c. Heat loss due to moisture in coal.
d. Heat loss due to moisture from H_2 of coal.
e. Heat loss due to CO in flue gas.
f. Heat loss due to carbon in ash and refuse.
g. Heat loss in dry flue gas.
h. Calculate equivalent evaporation, lb. per hour from and at 212°F.
i Calculate developed boiler horsepower.

j. **Calculate cost per 1,000 lb. of steam if the coal costs $3.50 per ton.**

k. **Calculate cost per 1,000,000 B.t.u. absorbed, coal at $3.50 per ton.**

12. *Boiler Test Data:* Hourly quantities—steam 174,400 lb., coal as fired 16,500 lb., ash and refuse 1,650 lb.; temperatures, air and fuel 65°F., flue gas at boiler outlet 594°F., flue gas at economizer outlet 274°F., feedwater at economizer inlet 212°F., feedwater at boiler inlet 318°F., steam at superheater outlet 711°F.; pressures, lb. per square inch absolute—steam (boiler and superheater) 425 lb.; flue-gas analysis, per cent CO_2 14.8, O_2 3.7, CO 0.05; coal, Somerset County, Pa.; relative humidity of air 46 per cent; carbon in ash and refuse 10.0 per cent; quality of steam leaving boilers 99.3 per cent.

Calculate the following items of heat balance, B.t.u. per pound of coal as fired, and as per cent of heating value of coal:

a. Heat absorbed by economizer, boiler and superheater.

b. Heat loss due to moisture in air.

c. Heat loss due to moisture in coal.

d. Heat loss due to moisture from H_2 of coal.

e. Heat loss due to CO in flue gas.

f. Heat loss due to carbon in ash and refuse.

g. Heat loss in dry flue gas.

h. Calculate equivalent evaporation, lb. per hour, from and at 212°F.

i. Calculate developed boiler horsepower.

j. Calculate cost per 1,000 lb. of steam if the coal costs $3.50 per ton.

k. Calculate cost per 1,000,000 B.t.u. absorbed, coal at $3.50 per ton.

13. Data: Proximate analysis, *M* 2.77, V.M. 14.69, F.C. 73.47, A. 9.07; ultimate analysis, S 2.79, H 4.02, C 78.71, N 1.46, O 3.95; B.t.u. per pound, as fired (by calorimeter) 13,774; Coal burned, lb. per hour, as fired 26,000; ash and refuse, lb. per hour, 3,197; 26.25 per cent of ash and refuse is unburned carbon; Flue-gas analysis, CO_2 13.2, O_2 5.8, and CO 0.7 per cent.

Steam evaporated, lb. per hour 248,000; temperatures: feedwater 260°F.; steam leaving superheater 620°F.; room 85°F.; flue gas 530°F.; steam pressure, 320 lb. per square inch absolute; quality leaving boiler 98 per cent; relative humidity 55 per cent.

Calculate the following items as B.t.u. per pound of coal, as fired, of heat balance:

a. Heat absorbed by steam in superheater alone.

b. Heat absorbed by water and steam in boiler alone.

c. Heat absorbed by water and steam in boiler and superheater.

d. Heat loss in dry flue gases (use 13.5 lb. of gas per pound of coal).

e. Heat loss due to moisture in coal.

f. Heat loss due to moisture from hydrogen of coal.

g. Heat loss due to moisture in draft air (use 13 lb. of air per pound of coal).

h. Heat loss due to CO in flue gas.

i. Heat loss due to unburned carbon in ash and refuse.

Three parallel tubes are placed along the top of each tuyere and extend from a header below the ash pit up to a header above the stoker hopper. For the extension grates, which are stationary in this design, parallel tubes extend from the same header below the ash pit to a header in the plenum chamber. These tubes are in the coal, and the circulation of water is naturally from the lower to upper headers. The headers are connected to the main circulation the same as any water-wall headers. Severe tests of this water-cooled stoker have proved its ability to burn the coal with low fusing temperature, at

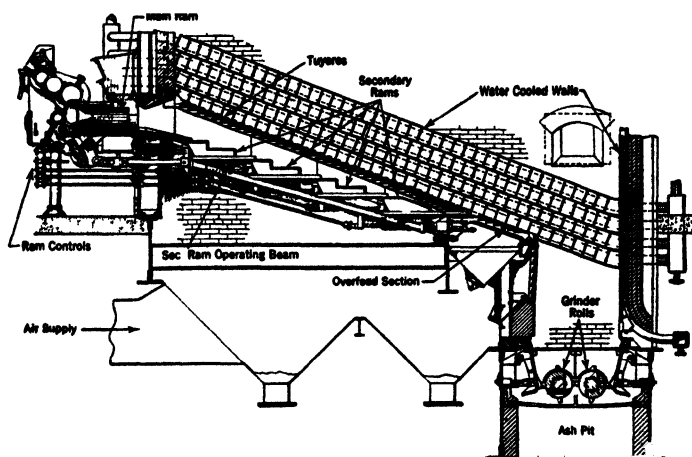



FIG. 71.—Sectional side elevation of CE multiple-retort stoker. (*Courtesy Combustion Engineering Company, Inc.*)

continuous rates of 48 lb. of coal per square foot per hour, of projected grate area, without damage.

The Combustion Engineering multiple-retort stoker, sectional view shown Fig. 71, has the moving parts in the retort bottom, and the sides of the retorts and tuyere blocks are fixed. The drive is either electric-motor or steam-turbine through a speed-change gear box.

The cylindrical main rams fit into flared openings in each retort, this design reducing the resistance to coal flow. The retort sides are built of cast iron and hold the cast-iron replaceable tuyere blocks. The depth of the retort is gradually reduced toward the overfeed section at the rear. Reciprocating auxiliary rams with sifting seals form the retort bottom. The travel of any auxiliary ram can be adjusted at the front of the stoker without stopping or interfering with the operation. The overfeed section is composed of alternately moving and fixed grate bars. Their function is to burn any combustible left at the end of the travel across the stoker and before the refuse enters the ash pit.

Zone control of the air is obtained by a series of dampers in individual ducts leading to tuyere stacks. These dampers are operated by hand by an operating shaft extending laterally under the stoker.

93. Overfeed Stokers.—Overfeed stokers are best suited for moderately small and medium-sized boilers. They usually use a natural draft and are particularly adaptable to burn low-volatile and

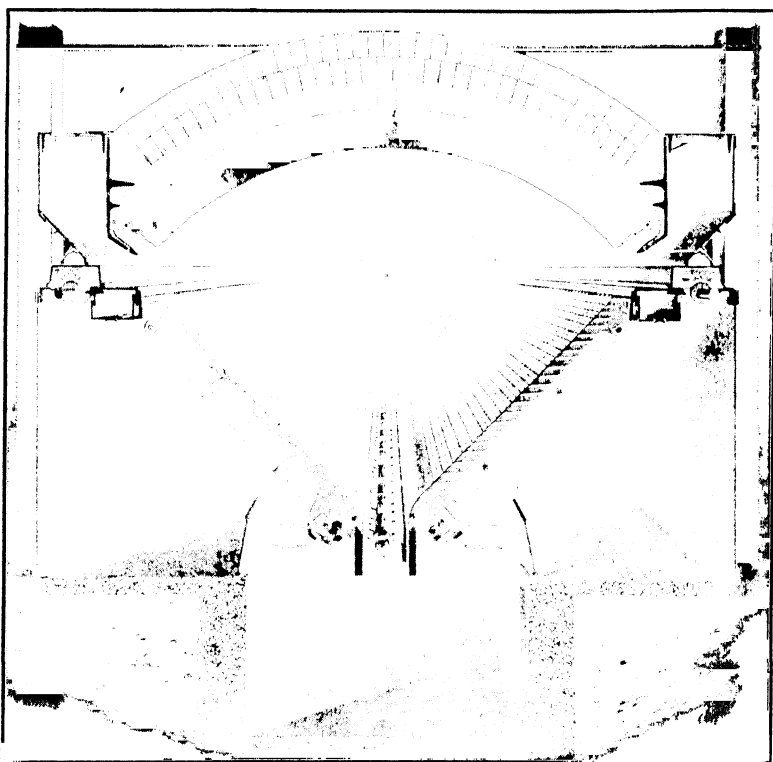



Fig. 72.—Murphy furnace and stoker rear-end view.

coking bituminous coals, but give reasonably good results with almost any fuel that will feed from a coal hopper.

There are two distinct types of overfeed stokers: the front-feed and the double-grate side-feed types. The former consists of one fuel hopper and a grate which slopes, at about 30 deg., to the rear of the furnace, while the latter has a fuel hopper at the top of each of two grates which slope from the sides, at about 45 deg., to the center and bottom of the furnace, forming a V. In either case fuel feeds from the hopper and is pushed by reciprocating stoker or pusher boxes onto coking plates where it is coked and ignited by the heat from

an overhanging arch. Aided by gravity, the fuel bed is "worked" to the bottom of the furnace by agitating grate bars, and the ash is deposited on hand-operated dump plates or fed into a clinker grinder.

In the *Murphy furnace* there are essentially two overfeed stokers that receive fuel from a hopper on either side of the furnace and dispose of the ash through a common clinker grinder at the center. In some

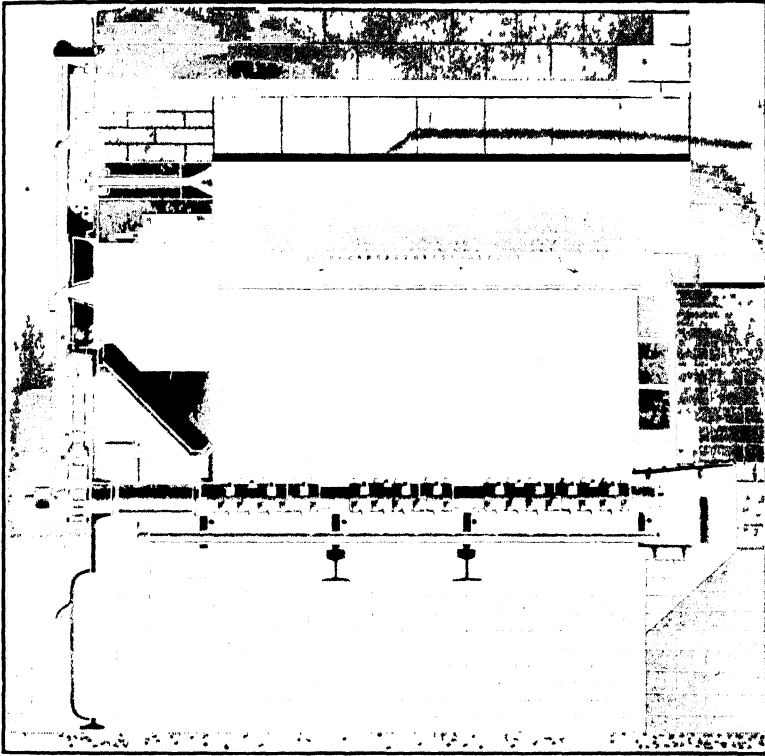



Fig. 73.—Showing sectional view of Murphy furnace and stoker.

installations the fuel is deposited on the top of the furnace and it is allowed to feed into the hoppers by gravity, while in others it must be supplied either mechanically or by hand from the furnace front.

Figures 72 and 73 illustrate the various construction features of the Murphy furnace. The grate bars are pivoted at their top ends and the bottom ends of alternate bars of each side engage with one of the two rocker shafts. The stationary bars rest on bearer bars that also serve as walls for the clinker grinder.

In operation, alternate grate bars rock about their top pivots, agitating the fuel bed, while the pusher boxes feed green fuel at the

top. The main air supply is through the grate, from the large chambers beneath. Additional air is preheated in the space above the furnace arch and flows through the series of tuyeres above the coking plates to the furnace. A circulation of air through the duct beneath the coking plates keeps them from becoming too hot. All the air is admitted through doors in the front wall of the furnace. The drive is usually by an electric motor or a small steam engine operating through suitable links to give the desired motion of the various parts.

COAL AND ASH HANDLING

94. Discussion.—The largest item of expense in any operating power unit is probably the combined first cost of the fuel and the cost of its handling. Consequently, it is highly desirable to place and build power plants so as to keep this portion of the total expense at a minimum. The location and construction factors, however, depend largely on the size of the plant and the nature of its service.

Because of the dominance of grate coal as an industrial power fuel, its handling, only, will be considered in the following articles. The handling of pulverized and other power fuels will be discussed in a later chapter.

In all plants, regardless of the size or whether they be hand- or stoker-fired, the completeness of the coal- and ash-handling equipment should receive careful consideration; the most elaborate systems are warranted in only the extensive power installations. A complete system may be divided into the following:

1. Unloading.
2. Storing.
3. Reclamation.
4. Crushing.
5. Delivery to bunkers.
6. Weighing and delivery to the stoker hoppers.
7. Disposal of the ash.

95. Unloading of Coal.—Coal is usually delivered to the plant by railway cars, barges or motor trucks. In any case the *method of unloading* should be such that a minimum of time and expense is obtained. When delivery is by rail, it should be in either the flat-bottom or hopper-bottom gondola car. From the former the coal may be unloaded by a crane and grab bucket, rotary car dumper or a pneumatic system. The rotary car dumper is perhaps the most modern method, and it consists of a means for holding and turning over the car sufficiently to completely dump the contents into a hopper below. Using hopper-bottom cars, the dumping action is accom-

plished by releasing hinged bottom sections of the car, which allows the coal to fall below to a track hopper or a trestle storage pile. If a portable conveyor is used, a track hopper or trestle structure is usually unnecessary.

Grab buckets, pneumatic vacuum tubes and portable loaders which can be handled by cranes are used, also, in *unloading coal from barges*. The grab bucket, operated from a steel crane or tower, is probably the most widely used method. The pneumatic system is often recommended for small-sized coal.

If coal is delivered by truck, as is frequently the case with isolated city plants, two types of trucks are commonly used. One is known as the end-dump type, which employs a tilting bed, and the other uses side or end gates and chutes. In any event, the coal-unloading facilities should provide the greatest convenience for dumping directly into the plant hopper or depositing in piles for storage.

96. Storing and Reclamation.—When possible it is often economical to provide for *storing* a reasonable quantity of coal. The seasonal market price of coal, car shortages, the weather and strikes are among the factors which hinder an economic and dependable fuel supply.

Aside from the expense involved, there is also the nature of the coal and its susceptibility to spontaneous combustion to be considered if storage is to be successful. It has been previously pointed out that lignites and some of the very soft coals deteriorate and slack considerably when they are exposed to the weather. Their heat loss, however, is surprisingly small. These same coals are also subject to spontaneous combustion if improperly stored. The exact cause of spontaneous combustion is unknown, but it is known that the percentage of volatile matter, the fineness and the accessibility of air are contributing factors. The moisture content and the height of the pile have considerable effect, also. Anthracite, however, has been stored under almost any conditions over long periods of time without any indication of ignition.

In general, coal may be stored successfully in silos, under water, on dry open surfaces under trestles, or in heaps forming continuous ridges, mounds and continuous areas, if the piles do not exceed 15 to 20 ft. in height. In any case care should be taken to avoid segregation of the lumps from the fine particles. Frequent inspection of the piles is necessary if there is any possibility of spontaneous combustion. The surest means of detection is, perhaps, that of inserting thermometers or thermocouples into the piles. With thermocouples the temperatures may be noted constantly from inside the plant. Temperatures of 150 to 160°F. would indicate that combustion is likely and that the pile should be moved.

The method of storing is largely dependent on the unloading and handling equipment, and there is a large variety of equipment available for this. The traveling and locomotive cranes with grab buckets and the drag-line bucket are probably the most common. Figure 74 shows a system of storage and reclamation using the drag-line bucket. The cables may be fastened to any of the steel back posts, to make spreading over the total storage area possible. The coal is dumped from the car and then crushed, after which it is raised by the elevator and dumped into the chute which deposits it in a pile near the

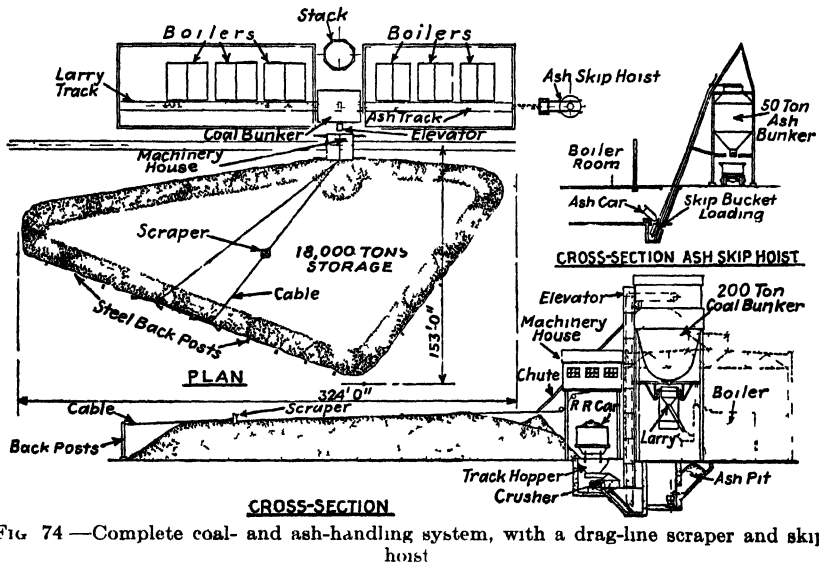



FIG 74 —Complete coal- and ash-handling system, with a drag-line scraper and skip hoist

machinery house. By means of the drag line and bucket it is carried to any part of the storage area. Reclamation is simply by scraping the coal in a reverse direction to a reclaiming hopper from which it is elevated to the overhead bunkers.

The storage system should provide for ease in reclaiming the coal. If it is stored directly from the cars, it has to be carried to the track hopper or deposited on a conveyor which will take it to the desired point.

97. Crushing, Delivery to Overhead Bunkers, Weighing and Delivery to Stoker Hoppers.—A plant seldom ever receives coal that does not have to be crushed either before or after storing to facilitate handling and burning. There are many types of crushers used for this purpose; the most common, perhaps, are those which pass the coal between a single roller and a stationary adjustable jaw or between

two adjustable rollers. Figure 75 shows, in general, the arrangement of the roller and jaw in a single-roll type of crusher.

Many plants use a breaker instead of a crusher. The breaker is a slowly rotating, nearly-horizontal cylinder, through which coal is fed. At the discharge end the broken coal, if fine enough, passes through a screen to a conveyor. The large lumps are returned to the entrance end of the breaker.

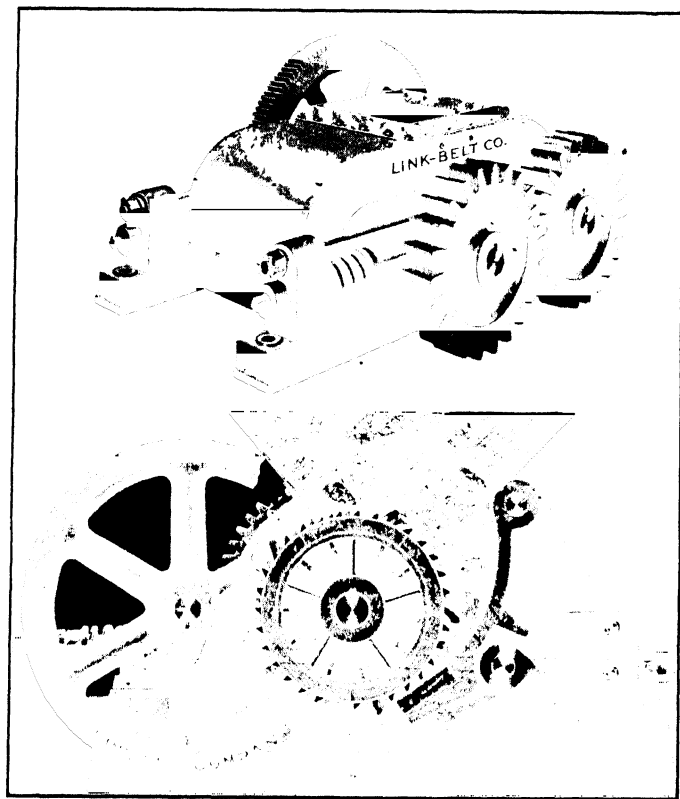



FIG. 75.—Single-roll coal crusher.

Some crushers receive the coal direct from the track hopper, but it is usually advisable to use some sort of an apron or a reciprocating feeder in order to properly regulate the supply and to avoid jamming of the crusher. In this connection a magnet device is often used to remove the tramp iron before the coal is crushed.

From the crusher or breaker the coal is usually delivered to overhead bunkers for brief storage, thus providing a ready and continuous supply for the stokers. To accomplish this a *continuous bucket con-*

veyor, a bucket elevator or a *skip hoist* is commonly used; the latter two usually in connection with a belt or flight conveyor running above the bunkers. Figure 76 shows the layout of a system employing the pivoted-bucket conveyor which serves for handling both coal and ashes. The buckets are carried between chains with wheels that roll on rails at the sides. The dumping devices in the upper run can be moved to dump the buckets at any point.

Another type of coal-handling equipment is the combination skip hoist and tram car, illustrated in Fig. 77. This system employs an automatic skip hoist for raising the coal to a receiving hopper above

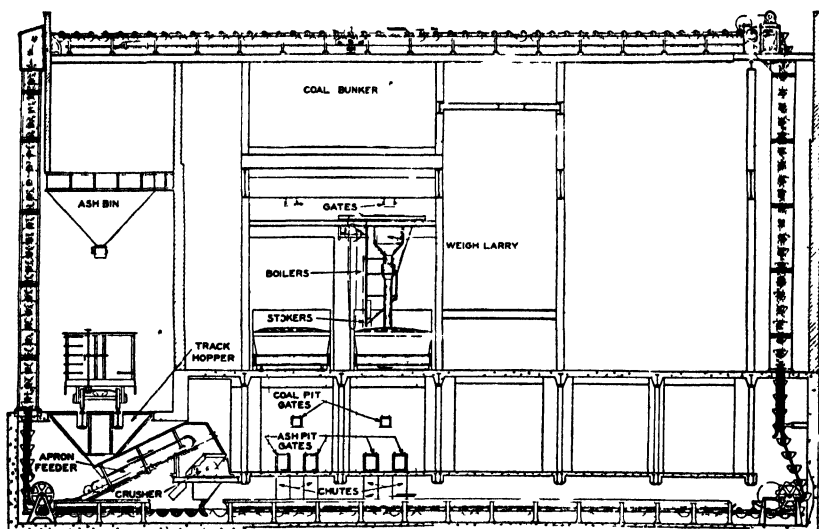



FIG. 76.—Complete coal- and ash-handling system, employing the Peck pivoted-bucket conveyor

the bunkers. Leaving this hopper, the coal is passed through a crusher, over a magnetic separator, and into a tram hopper which feeds the tram car. The tram car has the same capacity as the skip bucket and serves to distribute the coal to the bunkers. It is drawn back and forth over the bunkers in the same time cycle as the ascending and descending skip bucket and may be discharged at any point. The entire apparatus is operated by cables from a single winding machine, located as shown in Fig. 77. The skip-tram system is of recent design and has many distinct advantages over the older systems.

Coal bunkers are built of either steel or reinforced concrete, and of such shape that they can be practically emptied through their grates in the bottom. If a plant contains several boilers, one or more bunkers of considerable length are required.

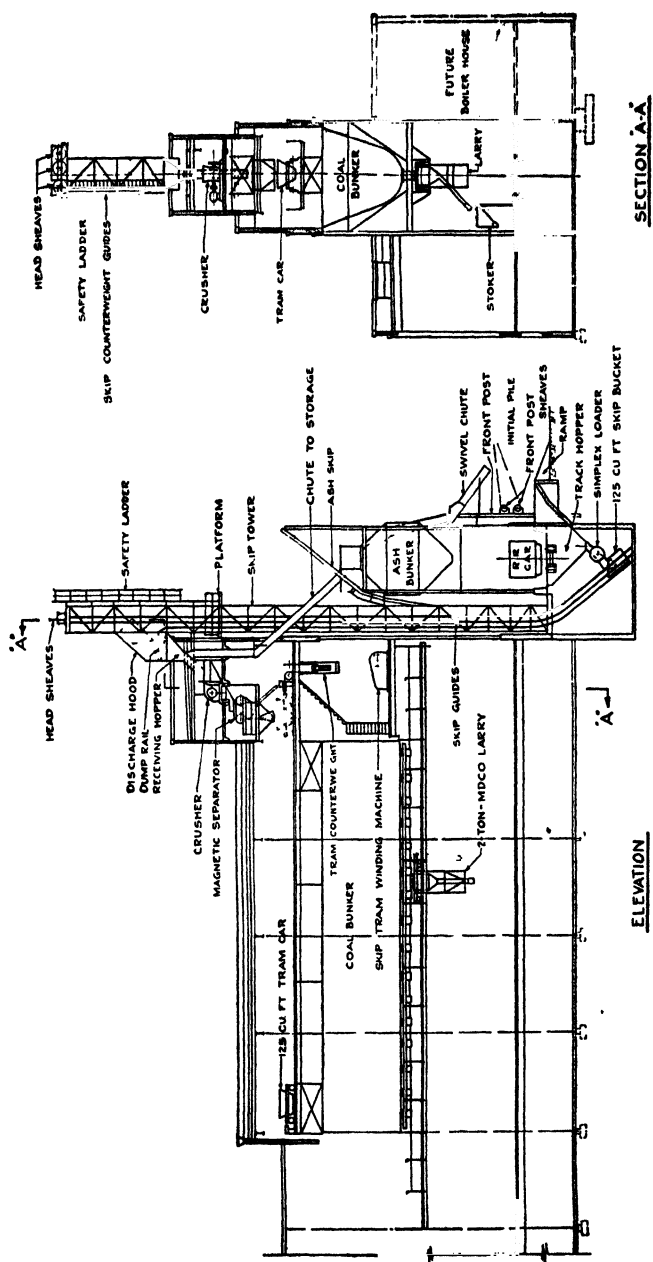



FIG. 77.—Beaumont combination tram-car and skip-hoist coal-handling system.

It is common practice to weigh the coal and to keep accurate record of the amount supplied to each stoker. Where this is done either stationary or traveling scales, located just below the bunkers, are used; otherwise the coal is discharged directly through spouts to the stoker hoppers. The traveling-type scale is often called a *weigh larry* and is usually operated from the boiler-room floor. By means of an electric motor or a hand chain, the larry may be moved to as to serve all boilers. Quite often, in the larger plants, automatic weight-recording devices are used to facilitate the weighing operation.

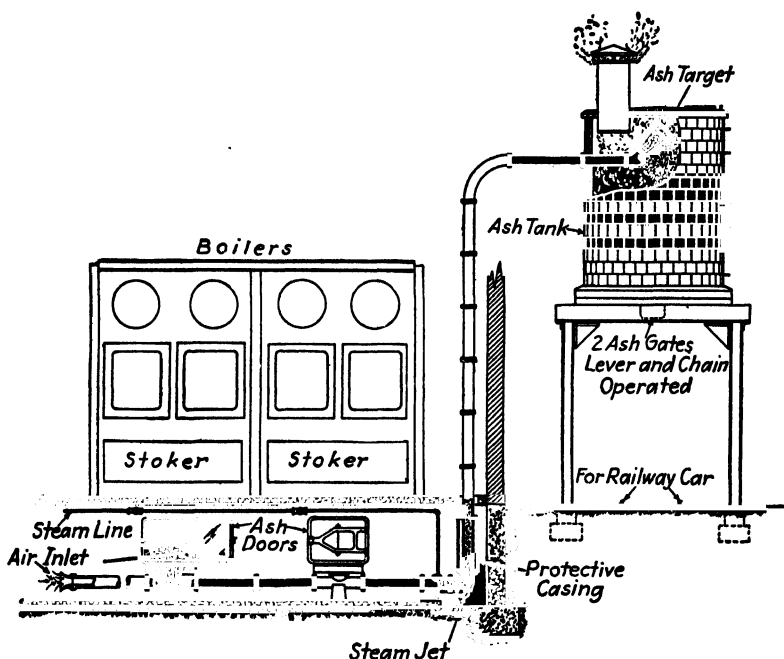



FIG. 78.—Steam-jet pneumatic ash conveyor. (*Design of United Conveyor Corporation.*)

98. Ash Disposal.—The methods and equipment used in handling ash and refuse vary over a wide range, depending, largely, on the size and characteristics of the plant. Frequently the same equipment is used for handling both coal and ashes (Fig. 76), but this is not recommended. Skip hoists, chain conveyors, pneumatic or hydraulic systems are often used in modern practice. There are plants, however, that are so arranged that the ashes may be discharged from a retaining hopper, beneath the stoker, directly into railway cars below. Figure 74 illustrates a typical system using the skip hoist.

One type of *pneumatic system* (Fig. 78) utilizes a steam jet to create an air flow through a conveyor pipe which carries the ashes from any

of the intake openings to a pile or tank outside the plant. The steam jet is located at the foot of a vertical riser, and the flow beyond the jet is by pressure alone.

Hydraulic systems employ an open or closed conduit containing a swift-running stream of water. The ashes fall by gravity or they are sliced or hoed into the conduits which carry them to a sump. The sump may be such that the water is allowed to drain off so the ashes can be removed by grab buckets, or it may contain a centrifugal pump which pumps the water and ashes into an overhead ash tank from which the water is permitted to drain off. The ashes are then dropped into cars or trucks, which are the final means of the disposal system.

Systems of ash handling incident to pulverized coal and other ash-containing fuels will be taken up in the following chapter.

CHAPTER VI

EQUIPMENT FOR BURNING PULVERIZED COAL, OIL, GAS AND WOOD

99. Pulverized Fuel Defined.—*Pulverized fuel* is taken to include all grades of coal that has been reduced to a powdered form for burning in suspension, as a gas, in an atmosphere of air.

100. Early Development.—The development of burning pulverized fuel in boiler furnaces dates from as early as 1876. The first attempts, however, were without noticeable success, chiefly, because of small furnaces. This difficulty was overcome, partly through coincidence, in 1895 when the Atlas Portland Cement Company adopted pulverized coal as a fuel for their cement kilns. About ten years later it was successfully applied to metallurgical furnaces, but it was not until about 1910 that any degree of success was obtained from burning pulverized fuel under steam boilers. Following this was a period of experimentation, which lasted about seven years; so it has been largely within the last decade that pulverized fuel, for the production of power, has acquired the degree of preference which it now holds.

101. Application of Pulverized Fuel to Power Boilers.— In general, all water-tube boilers and horizontal fire-tube boilers over 100-hp. capacity can be successfully fired with pulverized fuel. Many fire-tube boilers, however, are greatly handicapped by lack of furnace volume, but this difficulty is being overcome to some extent with the development of suitable burners. Recent experiments seem to indicate that a general application to Scotch marine boilers is not far off. Before the World War there was considerable agitation toward the equipment of locomotives for pulverized fuel burning, but this was suddenly stopped by the war and has not been attempted in this country with any enthusiasm since.

The many factors, such as ease and flexibility of control, high combustion and operating efficiency, adaptability of equipment to the burning of a large variety of coals and economy of banking, attainable with pulverized fuel, make its application to large installations, particularly, desirable. These advantages, however, are offset to some extent by the cost of preparation and the difficulty of ash disposal; the latter especially, when using coal the ash of which has a

high fusion temperature. The adaptation of pulverized fuel to power boilers has resulted in not only the introduction of a new and efficient method of burning coal but it has created an intense competition for stokers, which in turn has caused a general stimulation of enthusiasm in the whole field of combustion. Many of the advantages commonly attributed to pulverized fuel, such as water-wall furnaces, have been applied to stokers with remarkable success.

102. Preparation and Burning of Pulverized Fuel.—The preparation of coal for burning in the pulverized form consists of drying, pulverizing in a suitable mill, and supplying it to the boiler furnace with air, in a mixture suitable for burning. Part of the air for combustion is supplied with the fuel, as a conveying medium, and this portion

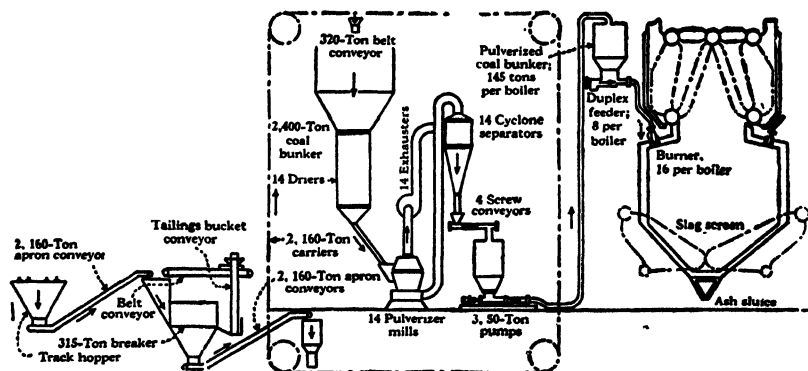



FIG. 79.—Diagram of a complete storage system for pulverized fuel.

(10 to 35 per cent) is called *primary air*. The remaining portion is called *secondary air* and may be supplied at the burner nozzle, or through suitable openings in the furnace walls, or both. The former is more recent and probably the better method. Advancement in both methods of firing is now taking place rapidly, and, for this reason, the advantages of one system over the other cannot be dealt with unreservedly.

There are essentially two systems for the preparation of pulverized fuel. One is known as the *central or storage system* and the other, the *unit or direct-fired system.* The central or storage system, shown diagrammatically in Fig. 79, receives crushed coal from driers and pulverizer and, by means of air or screw conveyors, delivers it some distance to bunkers near the furnaces. Such a system is adaptable, chiefly, to large plants where large and extensive equipment is warranted.

The *unit system* is gaining in favor, particularly for small installations. It consists of a pulverizer which receives raw (and in some cases dried) coal and delivers it through ducts to the burners in pow-

dered form and with a quantity of primary air. With raw coal the air supplied to the pulverizer is usually preheated to aid the reduction. Figure 80 illustrates a modern direct-fired installation using six burners and three pulverizing mills per boiler. The storage system gives the best distribution to burners and more flexible and better control, while the unit system overcomes the hazard of spontaneous combustion in storage bins and is also much simpler.

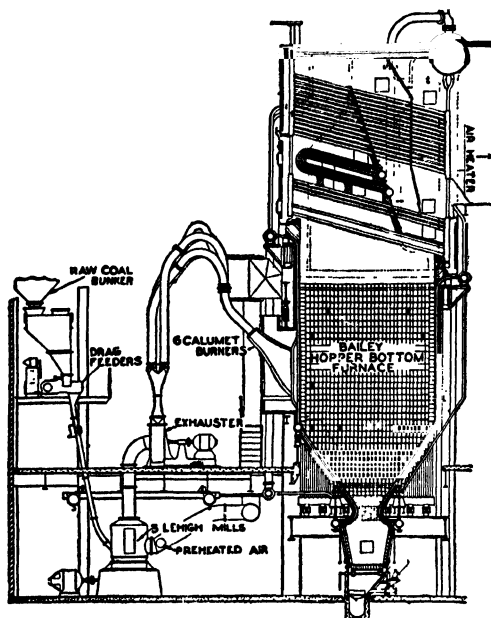



FIG. 80.—Direct-fired pulverized-fuel unit.

Generally speaking, coals high in volatile matter and low in ash content are best suited for burning in powdered form. These include all bituminous and lignite coals. Nevertheless, powdered anthracite is successfully burned, but because of its high percentage of fixed carbon it should be in a much finer state so the particles may be completely coked and burned before leaving the combustion chamber. The degree of coal pulverization or fineness is usually expressed as the percentage that will pass through a standard 200-mesh (openings per inch) sieve, though other sizes are used to some extent. Common practice considers that 70 per cent through a 200-mesh and all through a 50-mesh sieve is sufficient fineness in most cases.

Though fineness is highly favorable to combustion, other factors, such as dryness and turbulence in burning, have significant effects. The allowable moisture (2 to 20 per cent) depends largely on the

percentage of fixed carbon in the coal; for example, anthracite should contain very little moisture while lignite may run even higher than 20 per cent without extremely poor results. Turbulence and mixing of the coal and air are effected by the burner design, velocity of mixture leaving the burner and the method of introducing the secondary air.

103. Pulverized-fuel Furnaces.—Pulverized-fuel furnaces are, by necessity, much larger in volume than those of stoker-fired installations. The shape and size are determined, mainly, by the type of burner and the time required for complete combustion of the fuel. The time element is affected by the kind and nature of the coal, including moisture content and degree of fineness. The early furnaces were refractory lined and were fired with vertical burners. These were superseded, to some extent, by horizontal burners and furnaces having air-cooled walls which permit a much higher heat liberation (B.t.u. per cubic feet of furnace per hour). Under high-combustion rates, horizontal burning caused washing and erosion of the walls, and water-cooled walls (Fig. 45) were finally adopted. The present tendency is toward greater turbulence in burning, which is resulting in smaller furnaces and higher rates of heat liberation (20,000 to 35,000 B.t.u. per cu. ft.).

104. Ash Disposal.—A large portion (about 60 per cent) of the *ash* from pulverized fuel is carried through the boiler passes and out the stack, and that which remains is deposited at the bottom of the furnace from which it is removed at intervals. In some (slagging) furnaces, see Fig. 44, the ash is allowed to remain in a molten state, in which case it is tapped off, cooled, and carried away by water; in others (Figs. 40 and 127, pages 100 and 221) a water or slag screen is used to cause solidification before the ash reaches the bottom of the furnace. The removal of the solid or granular ash is by the ordinary means used with stokers. To avoid harmful effects, the ash which escapes with the smoke and gases is often, to some extent, recovered. Perhaps the most successful method used for the recovery is electroprecipitation, which precipitates the dustlike particles on electrically charged electrodes.

105. Pulverized-fuel Burners.—The design of pulverized-fuel burners is at present advancing with unusual rapidity. The tendency seems to be toward the induction of greater turbulence to reduce the furnace volume and render more complete combustion, greater capacity, and more efficient and flexible control of fuel and air.

The completeness of combustion depends largely on the degree and rapidity of mixing of the air and fuel. Mixing, in some burners, is effected by a long flame path, while in others, a turbulent action

generated in the burner, gives the desired result. Burners are usually classified on this basis, as follows:

1. Line-flow type.
2. Turbulent or flare type.

Burners of either type give excellent results if properly installed and

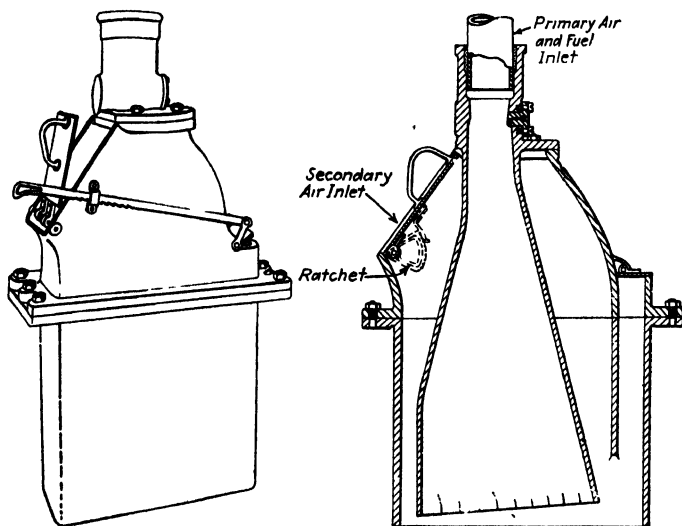



FIG. 81.—"Lopulco" natural-draft pulverized-fuel burner.

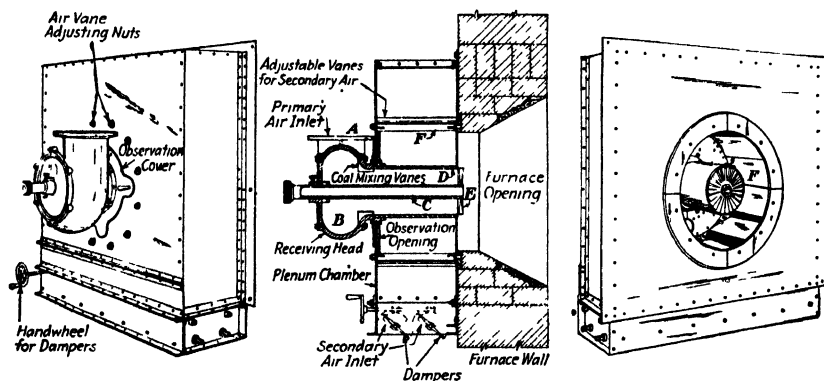



FIG. 82.—Riley flare-type burner.

operated, but it is generally considered that higher rates of heat liberation, giving higher furnace temperatures (2400 to 2700°F.), are attainable with those of the turbulent type. Turbulent burners are most frequently installed for horizontal burning, while those of the line-flow type are more adaptable to vertical burning, as this method requires a long flame and a deep furnace.

In operation, most burners receive the mixture of fuel and primary air (10 to 35 per cent of air required, at 5 to 20 in. of water pressure), mix with it the desired amount of secondary air, and discharge the whole mixture into the furnace for burning. The velocity of discharge must be regulated to exceed the velocity of flame propagation, which ranges from 15 to 45 ft. per second, depending on the quality of the fuel, its percentage of volatile matter, ash content, fineness and the

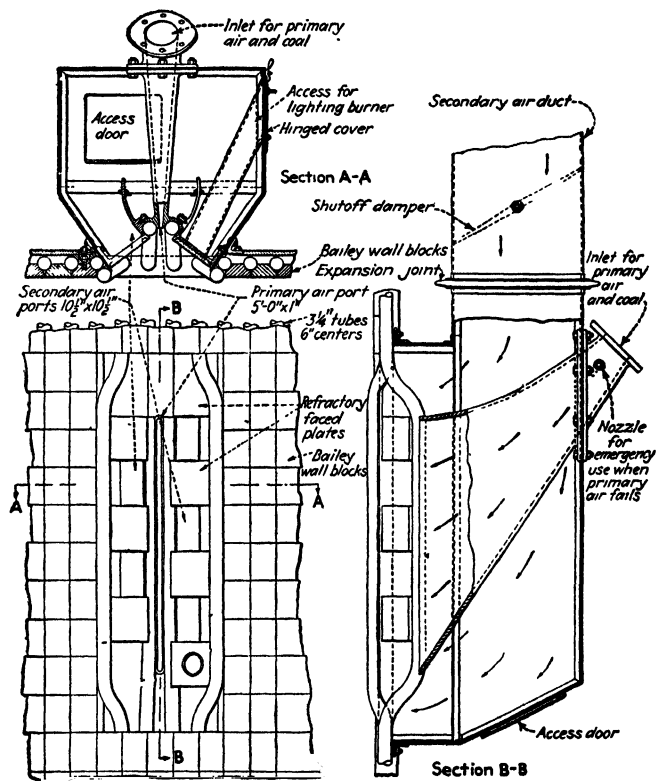



Fig. 83.—Calumet pulverized-fuel burner.

turbulence in burning. If the velocity is too great, the flame will "blow out," and if too small, it will "back fire" into the burner. The most desirable velocity is that at which the flame starts just beyond or at the mouth of the burner. Complete regulation is at all times under the control of the operator and, if properly handled, is given constant attention.

There are many good burners on the market, all operating on either of the aforementioned principles. Their capacities range from 1,500 to 10,000 lb. of coal per burner per hour, depending on size and whether

or not they are operated with natural or forced draft.* The burner illustrated in Fig. 81 is of the line-flow type and is built to operate on natural draft. Usually, there are several per furnace and they are mounted vertically, in a horizontal fire arch, as shown in Fig. 98 (page 184). The fuel and primary air are fed to each burner from a unit mill or separate feeder, through the small vertical pipe, and are blown into the furnace in a flat stream from the narrow fan-shaped nozzle. The secondary air enters through the openings in the outer casing, and the amount may be regulated by the door covers. *Lopulco* burners are also built to operate on forced draft and preheated air.

The *Riley flare-type burner* (Fig. 82) is built in all capacities and is typical of many of the modern burners operating on the turbulent principle. The fuel and primary air are fed through the opening *A*, on the exterior of the wind box, to the chamber *B*. The mixture then

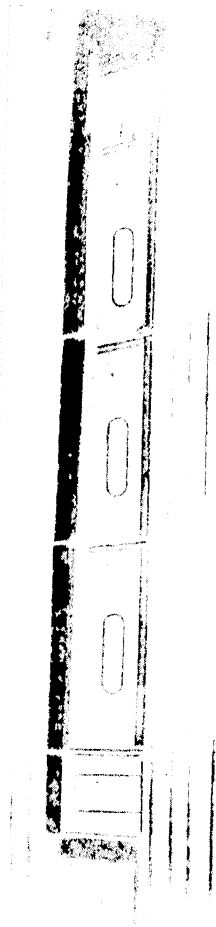



FIG. 84*a*.—Arrangement of burners for corner firing.

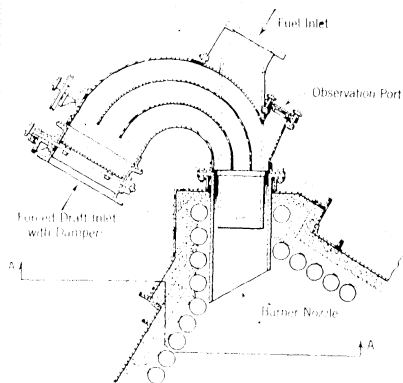



FIG. 84*b*.—Sectional-plan view of burner for corner firing. (*Courtesy Combustion Engineering Company, Inc.*)

passes through the space between the pipes *C* and *D* and is discharged into the furnace after being stratified and flared by the adjustable rosette vanes *E*. Secondary air enters the wind box at the bottom and is discharged into the furnace through the space between the adjustable spiral vanes *F*. The result is an extreme turbulent action radiating from the mouth of the burner. Initial ignition is secured by

using kerosene. The kerosene is first allowed to flow into the furnace through the small pipe *C*, after which it is ignited by an oil-saturated waste torch. Only fuel and primary air are first admitted. When the flame is sufficiently large the secondary air dampers are adjusted to give the desired amount of secondary air. If there is more than one burner per boiler the additional ones are ignited automatically on admitting fuel.

The *Calumet burner*, illustrated in Fig. 83, injects the primary air, carrying the powdered fuel, horizontally into the furnace through the long, narrow, vertical slot between the water-wall tubes. The secondary air enters at an angle through the alternate ports on either side of the primary air jet. The resulting action is a tearing of the fuel sheet, which causes a great amount of turbulence and excellent mixing of the fuel and air. This burner is built in capacities up to 8,000 lb. of coal per hour.

Figure 84 illustrates the corner-fired turbulent burner of the Combustion Engineering Company, used in their corner tangential-fired furnace (see Fig. 40, page 100). Fuel with primary air, and secondary air from different nozzles, join in the burner, producing a very turbulent flow into the furnace. This burner is adaptable for burning gas as well as coal.

106. Pulverizing Mills.—The mills used in reducing coal for burning in the pulverized form are essentially the same as those used for pulverizing other substances. Special design, however, has made them more adaptable to coal.

They are generally divided into three types, depending on their method of reduction, as follows:

1. Roller.
2. Impact.
3. Ball mill.

The roller-type mill crushes the coal to fineness with heavy steel balls or rollers which travel in a chilled cast-iron or hardened-steel race. Impact mills employ paddles or blades which rotate at high speed (900 to 1,800 r.p.m.). The impact of the blades and the attrition of the coal granules on one another effect, rapidly, the desired pulverization. A ball mill consists of a horizontal steel cylinder containing a large number of small steel balls. In operation, the cylinder rotates, causing a constant shifting of the balls which results in finely crushing the coal that is supplied at a constant rate.

Crushed coal is fed to mills by a feeder device, and after being pulverized it is removed by a flow of air induced by a fan which, in many cases, is integral with the pulverizing mechanism. The degree

of fineness of the discharged fuel depends largely on the velocity of the air passing through the mill. Consequently, many mills are built to utilize the lifting power of the air to select and remove the sufficiently fine particles as rapidly as they are formed. A separating screen is sometimes provided for the same purpose. In direct-fired installations the *primary air* is that which passes through the mill.

The mills used in connection with the storage system are generally of large capacities, ranging from 10 to 50 tons of coal per hour, while the unit or direct-fired mills are built in sizes to pulverize from 250 to 10,000 lb. of coal per hour. Many unit mills are equipped to operate with preheated air, in which case all other coal-drying apparatus is omitted.

A new design of pulverizer (Fig. 85) is the Raymond bowl mill by the Combustion Engineering Company. This mill is made in capacities ranging from 1½ to 15 tons per hour, or sizes from 34 to 54 in. The larger mills have three rolls, and the smaller ones two rolls.

The unique feature of the mill is a revolving bowl or grinding chamber. Due to centrifugal force the coal, which is ground between the reduction rolls and the grinding ring, works its way up the walls of the bowl. As the ground particles reach the rim, the fines and intermediate sizes are picked up by the air current coming up from the annular spaces around the bowl and are carried into the separator above for further classification. As the finished material is separated out, the oversize particles drop back and reenter the mill with the raw feed.

The rolls are held in a fixed position by the heavy journal heads supported on trunnions that rest on the top plate of the mill. Clearance between the rolls and the revolving bowl may be adjusted from the outside while the mill is running. The journal heads are connected through arms to roll pressure springs and adjustment rods mounted in cast-iron lugs outside the mill casing. By adjusting these compression springs the grinding pressure can be controlled to suit the requirements of the coal while the mill is running.

The bowl revolves on a vertical shaft mounted on roller bearings and is driven through a worm gear by a horizontal main shaft, direct-connected to a constant-speed standard motor. The vertical shaft and bearings are outside the grinding compartment and are lubricated by a continuous circulation of oil.

The mill parts are protected from damage resulting from tramp iron or ungrindable material by a positive means of disposal. The pressure springs allow pieces of iron to pass the rolls without injury, and when the tramp iron reaches the top of the bowl it is thrown into

the annular space below and expelled by revolving plows through a discharge.

For coal grinding and firing, the bowl mill is equipped with an air separator and automatic feeder, both mounted on the top plate. The exhaust fan is driven from an extension of the horizontal main shaft,

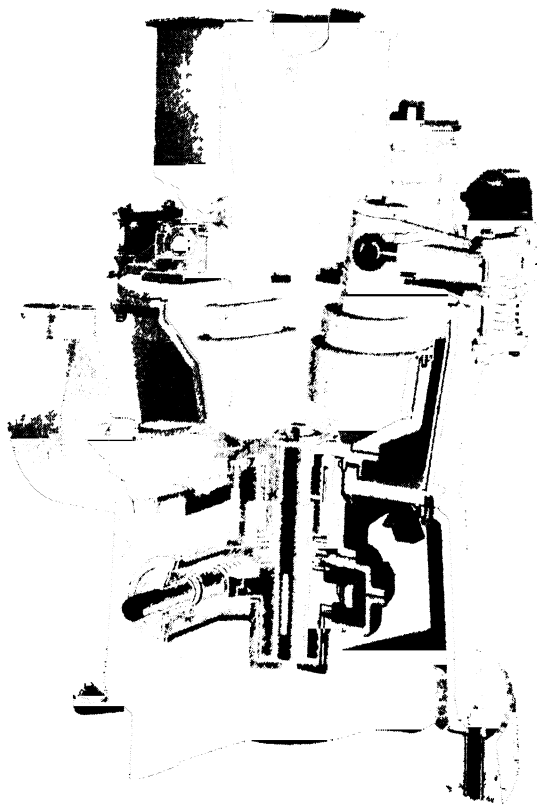



FIG. 85.—Section through CE Raymond bowl mill. (*Courtesy Combustion Engineering Company, Inc.*)

so that one motor drives the fan and the mill. Hot air may be passed into the grinding chamber when necessary for drying the coal.

Figure 86 illustrates the Type *B* Babcock and Wilcox pulverizer in sectional view. Coal is fed at an adjustable rate through the feed spout at the left. The rotating parts within the pulverizer include the vertical shaft and the attached ring between the upper and lower rows of balls. The rotation of this intermediate ring causes the balls to roll around their race, and the coal undergoes the process of pulverization.

The outlet to the exhaust fan is shown at the top of the casing and the return connection shows at the bottom and right. The compression of the top ring (stationary) is adjustable.

These pulverizers are built for capacities from 18,700 to 75,400 lb. of coal per hour, based on the pulverization of coal with a grindability[1]

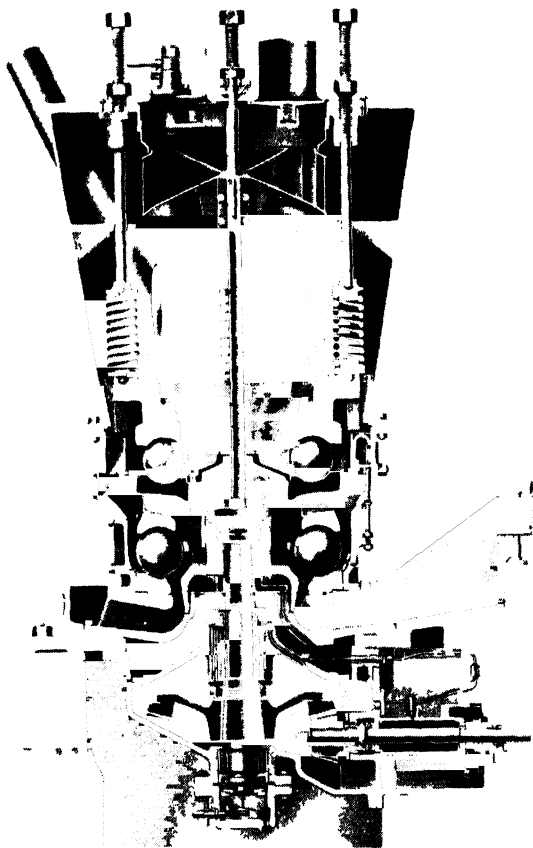



Fig. 86.—Babcock and Wilcox type B pulverizer.

of 57, to pass 65 per cent through a 200-mesh screen, and over 98 per cent through a 50-mesh screen. This type is used both for unit and storage systems.

The *Erie City Unitype* pulverizer (Fig. 87) is typical of the impact mills used for direct firing. Raw coal is automatically fed to the pulverizing chamber by the motor-driven feeder. Preheated primary

[1] For explanation of grindability see Grindability of Coal, by R. M. Hardgrove, *A.S.M.E. Trans.*, vol. 54, 1932, paper FSP-54-5.

air enters just below the feeder, sweeps through chambers and is discharged, laden with coal dust, by the fan, shown at the right end of the mill (Fig. 87). Pulverization takes place by gradual reduction in size, caused by the impact of the revolving blades or paddles on the granules as they pass through the three chambers or stages. The higher peripheral velocity of the larger blade wheel in the last stage effects the finest reduction. Additional air may be admitted to the fan through the small openings in the neck connection, between the last stage and the fan chamber. The Unitype pulverizer is built

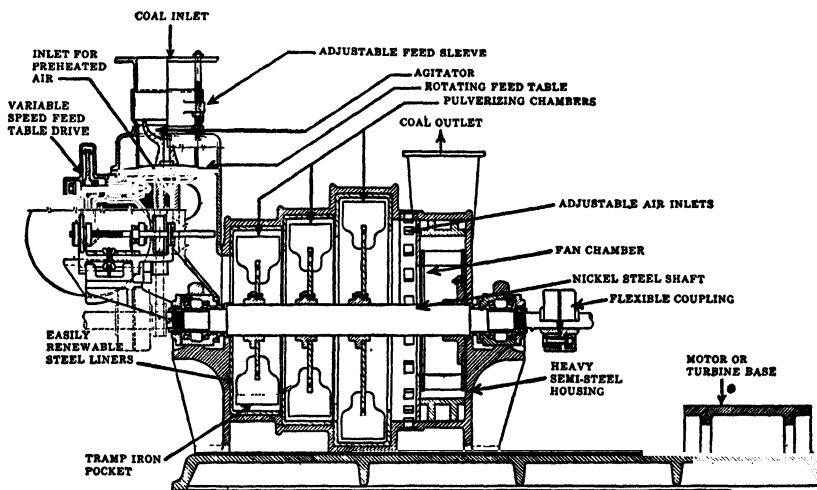



FIG. 87.—Erie City "Unitype" pulverizer.

in sizes to handle from 250 to 7,500 lb. of coal per hour and to run at speeds of 1,200 and 1,800 r.p.m., using the higher speed for mills smaller than 2,250 lb. in maximum capacity.

107. Coal Driers.—Drying of coal, as a preparation for pulverizing, is essential in many cases. The high-volatile coals usually require little drying, if fired direct, and this can frequently be accomplished in the mill, while coals low in volatility give considerable difficulty unless the moisture content is reduced to a low figure (1 to 4 per cent). In the storage system it is generally necessary, also, to dry the coal before it can be successfully conveyed and stored in the powdered form.

Driers commonly used may be divided into two classes as follows:

1. Rotary.
2. Stationary.

Both types are built to operate, using hot gases, preheated air or steam as the drying medium.

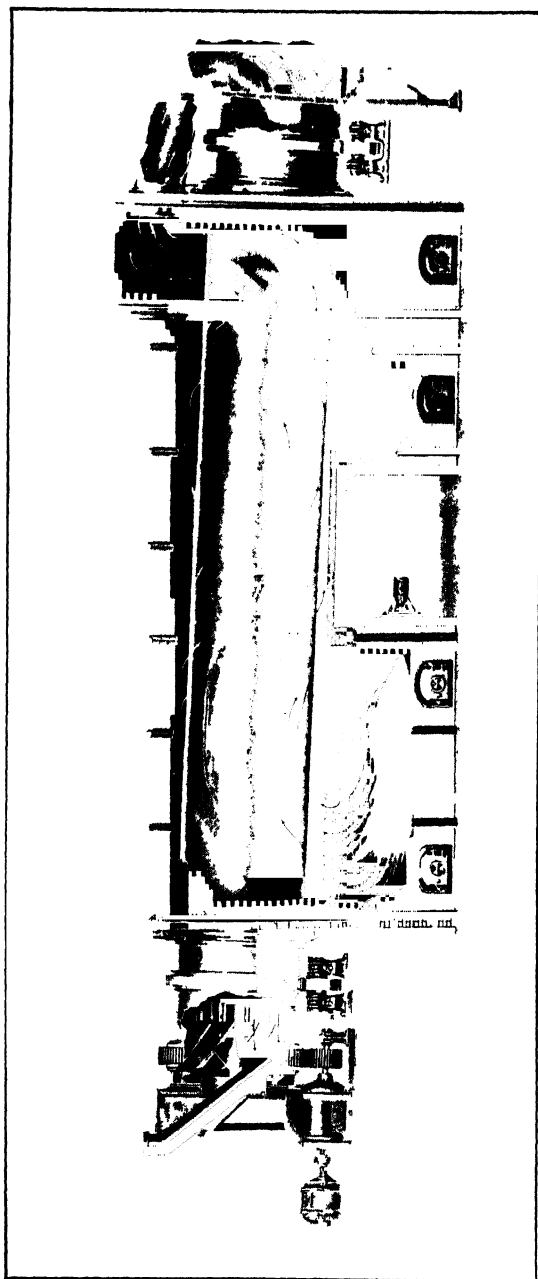



Fig 88.—Fuller-Lehigh rotary dryer.

The *Fuller rotary drier* (Fig. 88) is built in capacities of from 2 to 30 tons of coal per hour. Coal is fed to the inclined cylinder at the upper end and drying is effected by hot gases passing around and through it. As the cylinder slowly rotates (1 to 3 r.p.m.) the coal gravitates to the lower end and is discharged into a conveyor which carries it to the dry coal bunker. This drier is built to be fired with pulverized fuel. Others of the same type often use waste heat or crushed coal, oil or gas as fuel. The hot gases are drawn around the cylinder to the lower end, thence inside to the upper end, and out through the exhauster fan to a cyclone separator where any dust

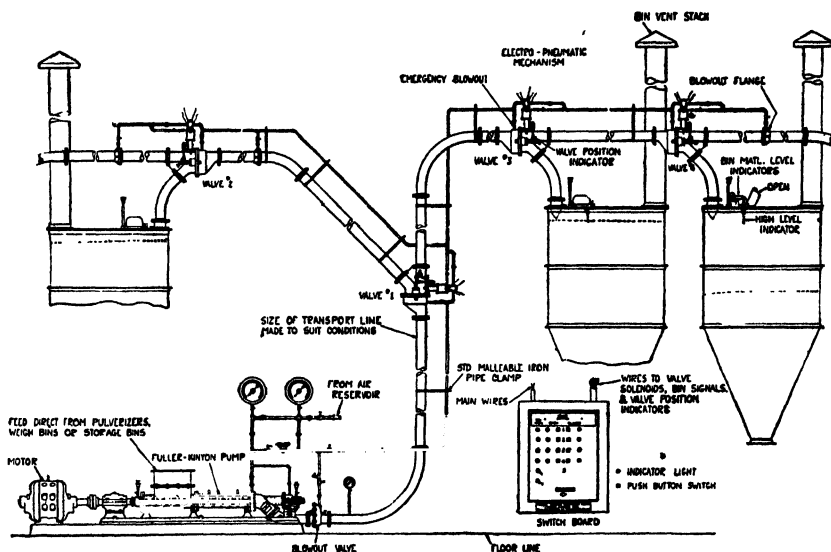



FIG. 89.—Fuller-Kinyon system for conveying pulverized fuel.

particles are removed. The space required for this drier renders it particularly adaptable to large installations where a central pulverizing plant is used.

108. Conveyors and Feeders.—Pulverized fuel may be conveyed by either a *mechanical* or a *pneumatic system*. In general, mechanical systems consist of either screw or especially constructed flight conveyors. They are best adapted to transporting over horizontal distances, such as between the collector and the storage bins, in a storage system installation. Over short distances, low-pressure air, produced by centrifugal fans, is used to sweep the fuel through large ducts to the burners or collectors. Systems which operate on high-pressure (5 to 9 lb. per square inch) air are frequently used for transporting fuel over long distances and up elevations, as is often

the case in storage-type pulverized-fuel systems. Figure 89 illustrates a complete and modern compressed-air system. The dried pulverized fuel is fed by gravity from a storage bin, weigh bin, or direct from the dust collectors to the screw pump (short screw conveyor). It is carried to the discharge end of the pump where the mass is aerated by a small amount of compressed air, thus changing the nature of the fuel from a composite mass into a semifluid, in which state it is transported through the conduit and valves to the individual service bins.

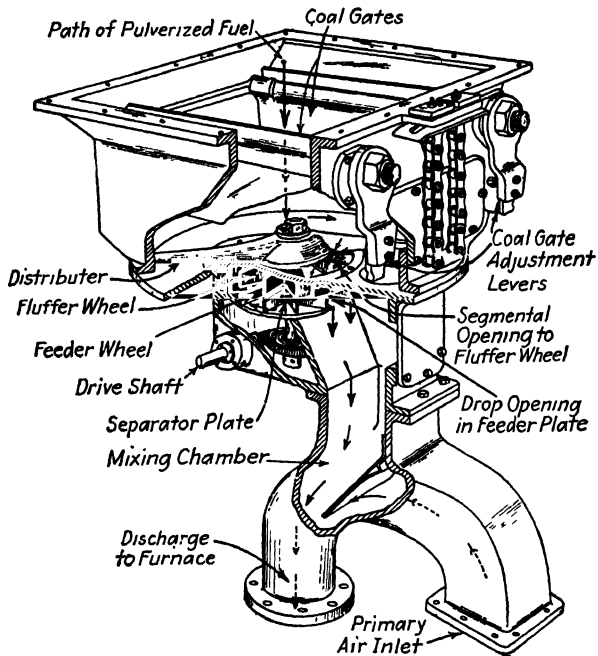



FIG. 90.—Bailey pulverized-fuel feeder.

By-passing the pump is an air line which furnishes air to the electro-pneumatic valves, at the storage bins, and, also, to the blow-out jets in cases of emergency. The system is provided with indicating or signaling devices, etc., and all distributing valves are operated from a common switchboard. The conduit is usually built of black steel pipe, ranging in sizes from 2½ to 8 in. in diameter, depending on the capacities required.

Feeder devices, as a part of a pulverized-fuel unit, are required for regulating the supply of granular coal to pulverizing mills and, also, powdered coal from bins to the individual burners. *Mill feeders* are generally built integral with the mills and in a large variety of designs. Their chief requirements are means for accurate and variable control

of the coal supply. This is especially true for direct-fired mills, as it is in this way, mainly, that the fuel supply to the burners is regulated.

Pulverized-fuel feeders play an important part in the operation of a storage system unit as it is by such means that the supply of fuel to the burners is controlled. There is a feeder for each burner and they are placed just below the storage or the weigh bin. The fuel is fed, at a definite rate, to the ducts carrying primary air and leading to the respective burners. There are several different makes of feeders on the market and in most cases they employ either a feed screw or rotating sectors. The latter is of more recent design and is often considered the more accurate, both as a feeder and as a meter.

The *Bailey feeder*, illustrated in Fig. 90, is well known and of comparatively recent design. Its moving parts consist of a vertical motor-driven shaft carrying an agitator or a distributor wheel and twin sector wheels below, known as the fluffer and the feeder wheels (Fig. 90). Between the distributor and the fluffer wheel is the floor of the feeder box. In the floor is an open sector which continues through an arc of about 150 deg. In operation, the powdered fuel feeds through this opening to the sector boxes of the fluffer wheel. Below the feed opening and between the fluffer and the sector wheels is a stationary division plate. The fuel is scraped off this plate to the feeder wheel which carries it around the bottom plate to an opening through which it falls to the mixing chamber. A flow of primary air then carries it to the burner. This feeder is driven by a variable-speed motor and it is built in sizes ranging from 4,000 to 10,000 lb. of coal per hour.

OIL-BURNING EQUIPMENT

109. General.—Oil, for steam-generation purposes, has many advantages over other fuels; among these may be included the following:

1. Ease of handling.
2. Convenience of storage.
3. Cleanliness.
4. Requires small amount of labor.
5. Permits good control and flexible operation.
6. Practically no ash.
7. Usually gives high boiler efficiencies.

On the other hand, oil is very often more expensive and is not so easily obtained as coal. And, also, while it seems to have a large number of advantages, the net operating costs, with oil, seldom fall below those obtained from the use of coal. Consequently, a general adop-

tion of oil, as a steam-power fuel, has gained but little headway in recent years. On shipboard, however, the first four of the advantages listed above are decidedly the determining factors. On one large Atlantic liner it was found that, on being converted to oil firing, the fuel-loading time was reduced to about 1/6 and the labor hours to about 1/120 of that required when using coal. There are also many other instances where oil is unquestionably the best fuel; it is largely a matter of the individual case.

110. Methods of Atomizing Oil for Burning.—To be burned under boilers, oil must first be atomized into a vapor-like mist, and then mixed with a sufficient amount of air for complete combustion. *Oil atomizers* are commonly called *oil burners* and consist of several types, depending on whether the atomization of the oil takes place with or without the aid of steam, air or gas. Burners of the former types supply the steam, air or gas at pressures of 25 to 100 lb. per square inch and effect atomization either inside or outside of the burner jet. The latter types are called *mechanical burners* and depend on supplying the oil to the burners under high pressure (100 to 250 lb. per square inch) to effect the desired spray.

111. Oil-burning Equipment.—In general, an *oil-burning system* consists of a suitable storage tank, oil pump, filters or strainers, one or more oil heaters, one or more burners, and the necessary piping. Oil-storage tanks may be built of either steel or concrete and are usually placed under ground, according to fire regulations. Concrete tanks are not recommended, except for heavy oils, when the seepage would be a minimum. It is often necessary to provide steam-coil heaters in or around the storage tanks, especially where extreme cold weather persists.

The *pumps* are of the direct-acting or rotary types, depending on the pressure necessary. With direct-acting pumps, an air cylinder to absorb the pulsations is often required. There is a large variety of rotary pumps used in oil-burning systems. The gear type is very common and gives good service with low pressures.

Oil-burner jets usually contain a number of minute or small holes, and, therefore, require oil that is thoroughly free from solid particles. To insure clean oil usually one or more strainers are installed in the system These are usually of the screen type, and are furnished with any desired mesh.

In operating oil burners it is necessary to supply the oil with a viscosity that will insure proper atomization. The temperature necessary to give the desired viscosity depends on the oil used, and it is rarely above 275°F. Excessive heating is not only wasteful but it causes

pulsations of the fire and admits an element of danger in case of leaks in the piping. *Heaters* used in preheating fuel oil operate on steam, and they are usually of the closed type, containing one or more straight or return tubes for oil passage. The steam used is practically negligible, and the condensate from it may be returned to the boiler.

Oil may be burned with either natural or forced draft. To obtain high boiler ratings forced draft is, of course, necessary, and this seems to be in keeping with the tendency of oil burning. The *furnace* construction depends, somewhat, on the type of draft used. In any case, however, a high-refractory fire-brick lining is required for maintaining a high furnace temperature, which is necessary to insure complete combustion of the oil vapor and gases before they reach the cooler heating surfaces. With natural draft, oil-fired furnaces are often built with provision to admit preheated air for combustion through checkerboard openings in the floor, just below the burners. The air is usually heated by allowing it to flow through ducts in the walls or floor of the furnace. Practically all forced-draft, and some natural-draft, units admit the air through large casings surrounding the burner nozzles. This method is usually the best, as it permits better control.

The burners are installed, usually, at the front of the furnace, so that the flame and gas travel are horizontal and upward to the boiler heating surface. In some cases, especially with small refractory furnaces, it is desirable to place the burners in the rear of the furnace so that the flame will travel horizontally forward, and then back and upward, before leaving the furnace proper. The main point to be considered is that the flame should not impinge on the furnace walls or any portion of the boiler heating surface.

112. Steam Atomizing Burners.—These are the most common of any of the non-mechanical types that are used for steam generation, and they have the advantage of requiring a minimum amount of equipment. The oil is supplied at low pressure (5 to 30 lb. per square inch), and the steam is always available, except, occasionally, when all boilers are cold. Very often a small steam generator or compressed-air unit is installed for starting purposes. The steam used by the burner is less than 2 per cent of the boiler output, and unless the loss of water is an objection, the operating cost is comparatively low. The adjustment and regulation of the steam and oil is by suitable valves near each burner. There are many good burners of this type available.

The *Enco burner*, illustrated in Fig. 91, is well known, and it is built to mix the steam and oil inside the nozzle. The steam enters through the upper connection as indicated. A portion of it flows

through the small Venturi nozzle, and the rest enters the mixing chamber through the small holes. Oil enters the nozzle from below and

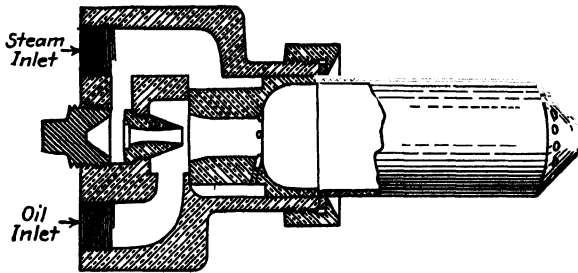



FIG 91.—"Enco" steam atomizing burner tip.

surrounds the small Venturi nozzle. The flow is then forward to the mixing chamber where the steam and oil are thoroughly mixed before

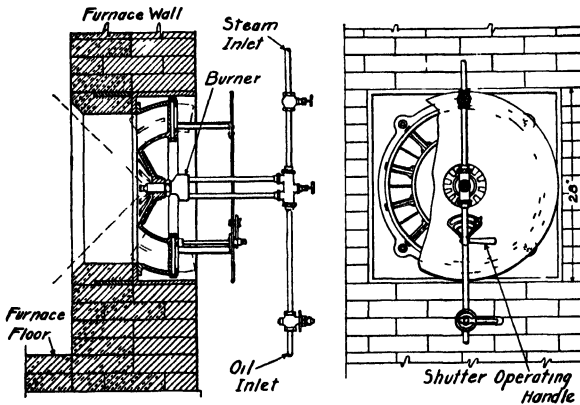



FIG. 92—Assembly of "Enco" burner and air register.

discharging through the end openings. The spray is conical in shape, which affords proper mixing with the entering air and produces a large flame. Figure 92 shows an assembly of the Enco burner with a

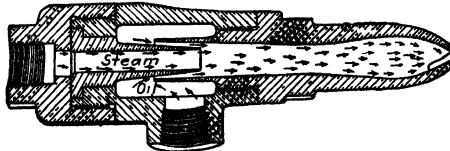



FIG. 93.—National "Airoil" steam atomizing burner.

natural-draft air register. It is also adaptable for use with forced draft.

The *Airoil burner* (Fig. 93) is also of the inside-mixing type. The entering steam flows horizontally through the small nozzle and picks

up the oil from around the end. The mixture continues to the mixing chamber where final and thorough mixing takes place before discharging at the end. The spray is somewhat fan shaped, and the discharge is directly into the flow of air for combustion.

The *Koerting locomotive burner* (Fig. 94) is an example of burners which effect atomization on the outside of the burner jet. The steam

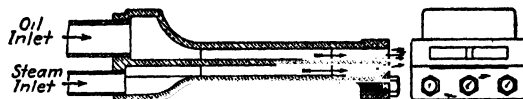



Fig. 94.— Koerting locomotive oil burner.

and oil flow as indicated and discharge in flat sheets at the nozzle end. The sheet of oil is directed into the steam and atomization results. This burner is usually installed at the rear and bottom of the locomotive firebox, so that the length of the flame, before reaching the tubes, will be a maximum.

113. Mechanical Atomizing Burners.—Burners of this type are commonly used in marine service where a loss of boiler water is highly objectionable. They require no direct steam for operation and are, therefore, conservative of feedwater. In operation, the oil is subjected to high pressure and is forced through a suitable nozzle, which breaks it up into a mist. The spray is directed into the path of inflowing air, which results in the desired mixture for combustion.

Fig. 95.—Koerting mechanical atomizing oil-burner tip.

Figure 95 illustrates a widely used mechanical atomizing burner and Fig. 96 the assembly of same for forced-draft firing. Oil is supplied through the gooseneck connection and is given a high rotative velocity as it flows through the spiral channels in the burner tip. The final discharge is through the conical tip and the resulting spray is somewhat cone shaped, as shown. The gooseneck, carrying the burner, is a separate element and is attached to the oil line by a clamp. This provision renders the burners interchangeable and facilitates removal for cleaning, etc. The air for combustion enters through the vane openings surrounding the burner. The supply may be regulated by a device which acts on the vanes and determines their amount of opening.

GAS-BURNING EQUIPMENT

114. General.—From the standpoint of operation and efficiency, a high-heat-value gas is, perhaps, the best fuel attainable, for steam gen-

eration. Its cost, however, is in most cases prohibitive. The chief exceptions occur, usually, in the regions where natural gas is in abundance, or in industries which produce gas as a by-product. Also, many small industrial plants which have isolated steam units consider gas as most economical. It is often said that gas has all of the advantages of other fuels without any of the disadvantages, and for these reasons the primary cost is often of secondary importance.

115. Gas-burning Equipment.—Gas furnaces are usually equipped to burn gas with either pulverized coal or oil simultaneously, or as a stand-by fuel. A large combustion space is required in gas-fired

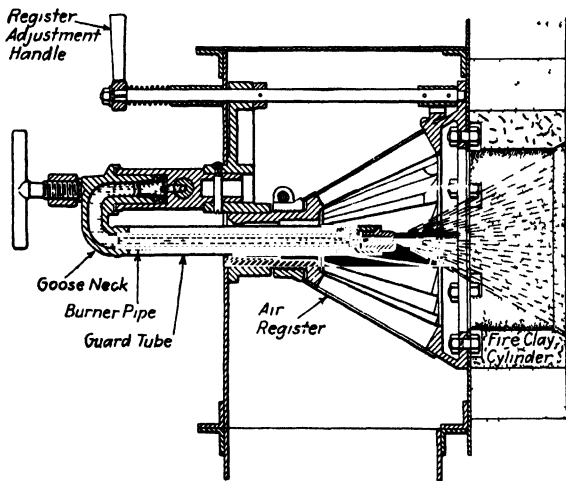



Fig. 96.—Showing assembly of Koerting mechanical atomizing burner, with forced-draft air-control register.

installations, generally not less than 1 cu. ft. per developed horsepower at maximum load. Blast-furnace gas is ordinarily supplied at from ½ to 50 in. of water pressure, depending on the burner, while, in the majority of gas burners, the pressure ranges from 3 to 10 in.

The design of a gas burner depends, largely on the kind of gas. Some effect the mixing of the gas and the air for combustion inside the burner jet, while others are built similar to oil burners and admit the air to the furnace, outside and around the burner jet. The *Duquesne burner* (Fig. 97) is of the latter type and consists of a cast-iron box to which are connected a large number of nozzles. These nozzles extend toward the mixing plate and terminate just inside the large holes. The gas is regulated by a valve, near the burner, and it flows into the burner box through the flanged connection at the top. The discharge from all nozzles is substantially the same. Air enters the

burner through the adjustable opening, as indicated, and flows around the nozzles and into the furnace, with the gas, through the large holes in the mixing plate. The air supply is regulated according to the gas-pressure indicator at the front of the box. This burner operates on from 1 to 4-oz. gas pressure. The use of from one to ten burners per boiler, depending on the size, is a common occurrence.

Figure 98 illustrates the Combustion Engineering Company steam-generating unit at the River Rouge Plant of the Ford Motor Company. This unit, arranged to burn three fuels, 100 B.t.u. blast-furnace gas, 550 B.t.u. coke-oven gas, and powdered coal, and producing steam at 1400 lb. pressure and 900°F., is the world's largest

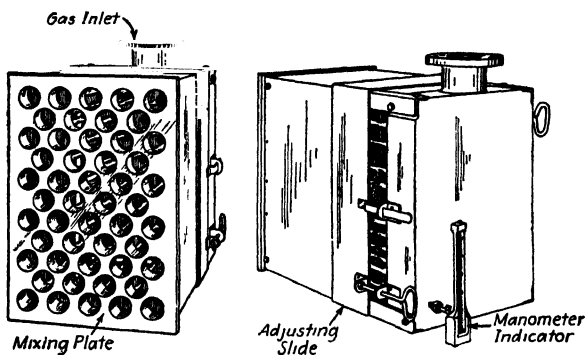



FIG. 97.—Duquesne natural-gas burner

high-pressure unit. A description of the equipment shown should be of interest.

In the illustration, one half of the boiler and equipment is shown in cross-section, the other half gives the external view. The Ladd boilers, vertical, bent-tube type, are set double over a common furnace. For the unit, the boiler heating surface totals 30,000 sq. ft. There are 616 3-in. tubes on 7-in. spacing, and 1,180 2½-in. tubes on 5¼-in. spacing. The larger tubes are located in the front bank next to the furnace, the greater space between tubes providing space for the tubes of the first superheater banks.

The four drums in contact with water have forged cylindrical shells, and welded thereto the pressed heads. Dimensions are as follows: two lower drums, 40 in. in diameter, 4 in. thick, and 30 ft. in length; two upper drums, 48 in. by 5¼ in. by 34½ ft. The dry-steam drum, a welded drum, centrally located above the setting is 40 in. by 3$^{11}/_{16}$ in. by 24½ ft.

Two convection-type superheater banks, in series, are located in each boiler side. The total heating surface of the four superheaters

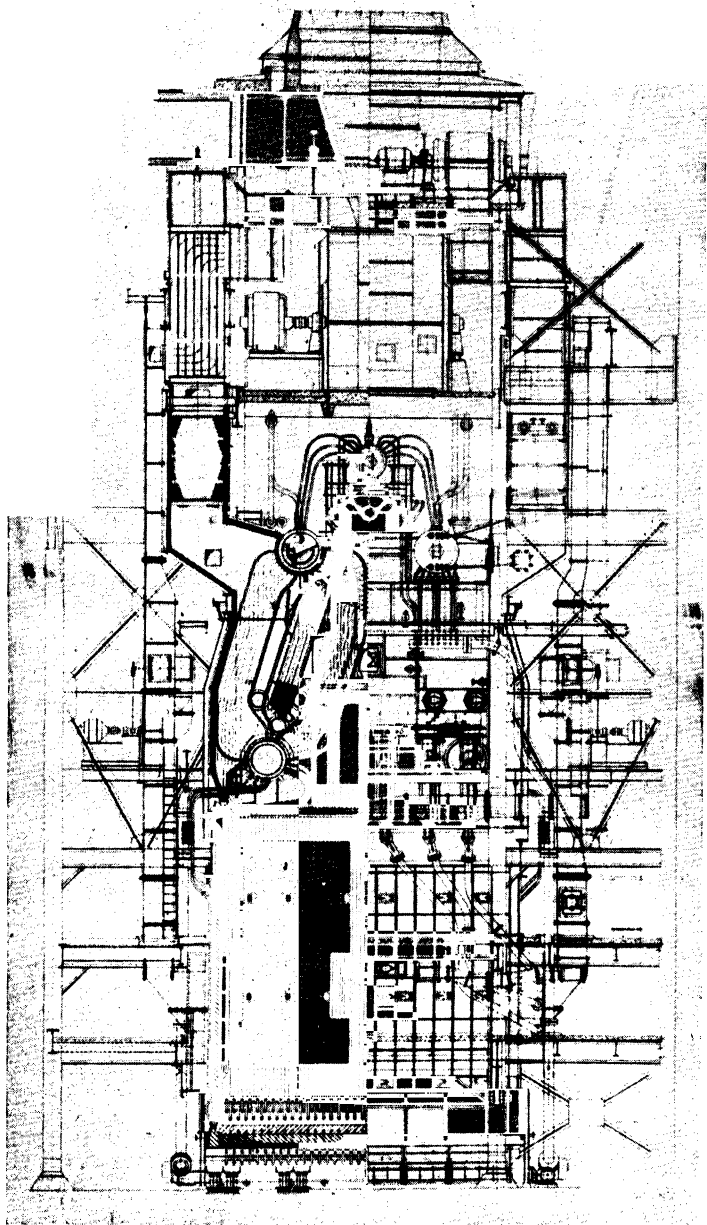



Fig. 98.—1400-lb. pressure steam-generating unit at River Rouge Plant, Ford Motor Company. (*Courtesy Combustion Engineering Company, Inc.*)

of the unit is 21,940 sq. ft. On each side, one bank of superheater tubes is placed between boiler tubes of the first gas pass, just back of the first row of tubes. Thus some radiant effect is provided. The second superheater bank is placed in the second gas pass, with a damper by means of which the gas flow can be largely by-passed around this element when the steam temperature becomes too high. Gas temperatures in the first boiler pass run from 2200 to 2300°F. producing 750°F. steam leaving the first element. In the second pass, gas temperatures of about 1600°F. bring the steam temperature to 900°F.

All furnace walls are protected by 3-in. fin-tube water walls, and water-cooled ash screens are over the hopper bottom. These tubes provide 7,200 sq. ft. of radiant-heat-absorbing surface, their water circulation being connected through headers into the boiler circulation.

The furnace with a volume of 30,000 cu. ft. burns pulverized coal as the main fuel, the other two being burned when available. In common practice blast-furnace gas is burned at an average pressure of about 30 in. of water and a maximum pressure of 48 in. of water. If the pressure drops to below 10 in. this fuel is shut off until the pressure builds up. Gas burners are located centrally in each front wall. Twelve pulverized coal burners are arranged in four sets of three burners, one set for each corner of the furnace. Each burner discharges its stream of fuel and air tangentially to an imaginary circle with its center the center vertical axis of the furnace. A very turbulent, whirling, vortex effect is produced in the furnace. The maximum boiler evaporation is 900,000 lb. steam per hour. The approximate fuel heat release is 34,000 B.t.u. per cubic foot of furnace volume.

The unit includes two fin-tube economizers with total heating surface of 26,500 sq. ft., and two vertical-plate, counterflow, 224-element, air heaters with total heating surface of 86,016 sq. ft.

The guaranteed efficiencies of the unit are:

Evaporation, Lb. per Hr.	Efficiency, Per Cent
400,000	87.7
600,000	87.8
800,000	87.4
900,000	86.6

WOOD-BURNING EQUIPMENT

116. Wood as a Fuel for Steam Generation.—In recent years the use of wood fuel for steam generation has gained considerably in importance. Like blast-furnace gas, wood fuel is generally refuse derived from an industrial process and for many years was considered a "dead

waste." In the lumber districts of the Pacific Coast there are now many power plants which burn wood exclusively. Where the supply of wood waste is insufficient, as in many woodworking industries, provision is often made to burn wood as an auxiliary fuel or in combination with other fuels.

117. Preparation for Burning.—The form in which waste wood is available depends, to some extent, on the nature of the industry. It may consist of sawdust, shavings, blocks, slabs and edgings in various mixtures which are unsuited for efficient burning. Consequently, it is often necessary to reduce these pieces to more or less uniform size to facilitate burning and handling. To do this they are put through a shredding or "hogging" machine which effects reduction to small chips or shavings, commonly known as "hogged" fuel. Figure 99 illustrates, in general, the construction and operation of a typical shredding machine. The hammers rotate at from 1,100 to 1,800 r.p.m., depending on the size of the machine. Shredding is accomplished between the hammers and the shredding bars or knives in the lower part of the machine. The discharge may be into a conveyor or a suitable bin. The shredding blades or bars are removable, and the type to be used depends on the nature of the wood and the fineness or degree of reduction desired. To produce sawdust a steel screen plate, containing a large number of small holes, is substituted for the bars.

118. Equipment for Burning Wood.—The type of furnace used for burning wood fuel depends, to some extent, on the moisture content and whether or not it is to be burned in combination with other fuels. For wood alone, the Dutch-oven type of furnace is probably the best. The long gas travel tends to afford more complete combustion of any wood particles that are picked up and carried away by the draft. A typical wood-burning furnace is shown in Fig. 36, page 96. The grates are sometimes slanted rather steeply toward the bridge wall, and occasionally an A-shaped grate, running longwise of the furnace, is found to be most satisfactory. Forced draft is seldom used with wood fuel, principally because of its light weight. Even with natural draft a large portion of the dry and finer particles is burned in suspension.

119. Storage and Handling of Wood Fuel.—To facilitate handling and storing it is ordinarily best to first reduce wood fuel to the hogged form. Storage may be either indoors or out in the open. When in the open a drag scrape is used for the handling. Inside the plant and for transporting over reasonably long distances, conveyors of the flight- or drag-chain type are commonly used. For efficient firing it is best to

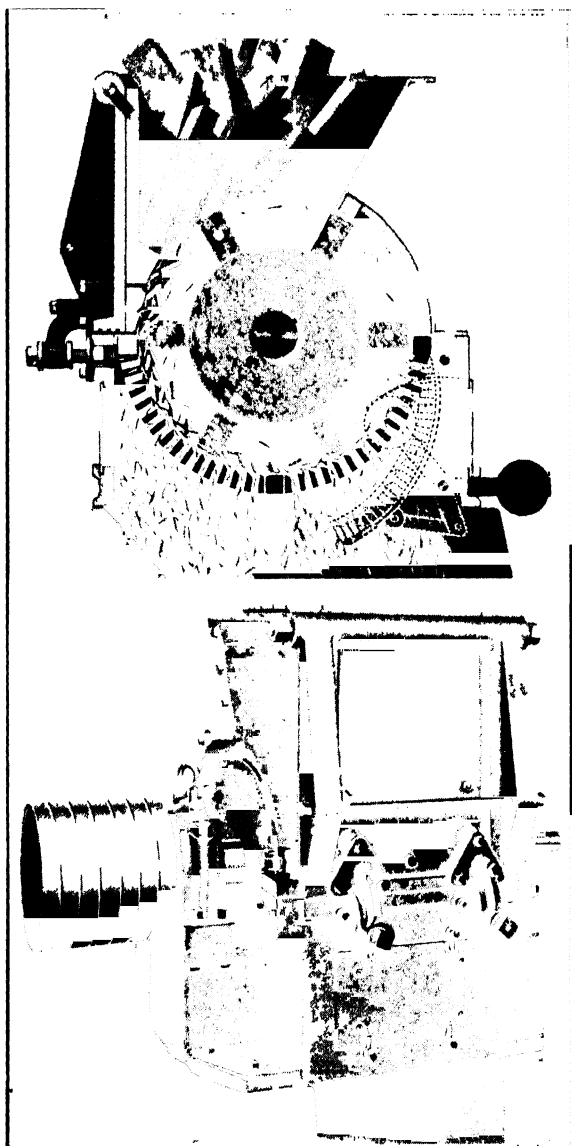



FIG. 99.—Jeffrey wood shredder.

feed the fuel to the furnace at a definite and constant rate, which is accomplished with especially constructed feeders. It is often recommended that hogged fuel be fired moderately wet so that it will stay on the grate while burning. When this is desired, a water spray is installed in the feed chute.

The unit of measure for wood fuel is usually the cord, which has a volume of about 100 cu. ft. and weighs from 2,500 to 4,000 lb., depending on the wood and its moisture content. Transportation over long distances is usually by special barges or gondola railway cars having a large volume.

CHAPTER VII

DRAFT EQUIPMENT, AIR PREHEATERS, AND COMPRESSORS

120. Definition and Cause of Draft.—In connection with combustion, *draft* refers to the difference in pressure available for producing a flow of air and hot gases. Such pressure differences are ordinarily small and are usually expressed in inches of water. Thus, in a boiler setting or chimney a draft of 1 in. of water signifies that the pressure of the atmosphere, as measured by a barometer, exceeds the absolute static pressure of the hot gases at the point of measurement by an amount sufficient to balance a column of water 1 in. high. If the passages are unrestricted a flow occurs, and it is in this way that air for combustion is supplied to the furnace, the hot gases circulated over the heat-absorbing surfaces of the boiler, and the waste gases discharged to the atmosphere.

Fig. 100.—Simple manometer draft gage.

121. Draft Gages.—Draft pressure, or the *intensity* of draft as it is sometimes called, may be measured by means of a water manometer, the simplest form of which is shown in Fig. 100. It consists merely of a glass U tube half filled with water, open to the atmosphere at one end and having the other connected to a pipe leading to the point in the boiler setting or stack where the pressure is to be measured.

Fig. 101.—Ellison inclined draft gage.

The difference in level, as measured on the accompanying scale, indicates the draft pressure in inches of water.

There are many types of *draft gages* in use. Figure 101 illustrates a type of manometer gage in which the pressure is read on a scale opposite a slightly inclined tube. Colored oil is the liquid used in

the tube. The scale is calibrated to read in inches of water. This style of construction greatly magnifies the pressure reading and permits the reading of very small pressures on a scale of considerable length.

Figure 102 illustrates a pointer type of draft instrument which consists essentially of four separate draft gages. The operation of this gage is readily apparent on referring to the sectional view. The pressure to be measured is exerted under a weighted bell which is inverted in a light-gravity oil. Variable pressures cause the bell to rise and fall, and this movement is transferred through a straight-line mechanism to the pointer. This instrument is built having one or more pointers and with scales for various pressure ranges.

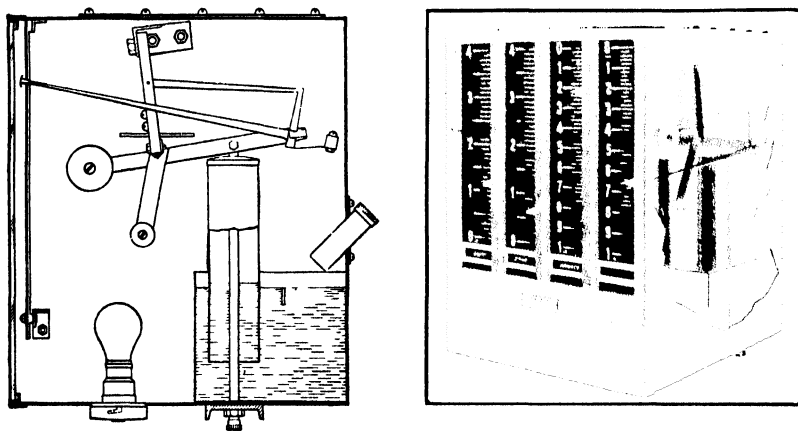



Fig. 102.—Ellison multi-pointer draft gage.

Occasionally it is desirable to obtain a continuous record of draft pressures. In such cases a recording gage is used.

122. Draft Losses.—Draft loss is taken to mean the reduction in measured draft pressure (inches of water) in all, or any part, of the boiler equipment, between the ash pit, stoker wind box, or burner box and the chimney; that is, all equipment through which the air or hot gases must flow.

The draft at the flue or outlet of a boiler setting must be sufficient to cause a flow of the gases and overcome the frictional resistance of the grate and fuel bed (or burner box), boiler passes and the damper. Other restrictions to flow exist in the gas passages between the setting and the chimney, and also in the chimney itself. Among these are frictional resistances offered by the straight portions of the ducts and chimney, and that due to abrupt turns and angle connections. If an economizer or air heater is used, each of these adds considerably toward restricting the flow of gases.

These draft losses vary in magnitude over a considerable range and depend chiefly on the fuel used, the rate of operation, the type of equipment used and the design and arrangement of the gas passages. For convenience of discussion and calculation, the various losses will be considered separately.

The *grate* and *fuel-bed* resistances are usually given as one item, and the greatest portion of this loss is through the fuel bed alone. The size and kind of fuel, and the thickness of the fuel bed, are the main influencing factors.

TABLE 7-1.—AVERAGE VALUES OF GRATE AND FUEL-BED RESISTANCE

Class of fuel and stoker type	Head, in. of water	Coal-burning rate, lb. per hr. per sq. ft. grate area
Hand-fired grate fuel:		
Anthracite	0.15 to 1.0	10 to 35
Bituminous, west	0.05 to 0.8	10 to 50
Bituminous, east	0.05 to 0.6	10 to 50
Chain grate:		
Bituminous, east	0.05 to 0.6	10 to 45
Bituminous, Rocky Mountains	0.05 to 0.45	10 to 45
Underfeed:		
Bituminous	1.0 to 6.0	10 to 90

As a general rule, the required draft increases as the percentage of volatile matter in coal decreases. Oil, gas, or pulverized fuel requires less draft than coal on the chain grate.

The loss through a *boiler* may vary between 0.1 and 1.5 in. of water, depending on the type, size, method of baffling and the rate of operation. Boiler loss is about as follows:

Inches of Water	Rating, Per Cent
0.1	50
0.3	100
0.7	200
1.9	300

A superheater adds, usually, about 0.15 in. to the loss through the boiler.

In the *ducts* and *breeching* a loss of 0.1 in. per 100 ft. of run and an addition of 0.05 in. for each right-angle turn is considered a reasonable approximation.

Steam gives the following equations for calculating draft losses in ducts (stacks, flues, and breeching), and through the economizer:

$$D_5 = \frac{C_1 w_g^2 PLT}{A^3} \quad \text{in. of water} \tag{81}$$

$$D_5' = \frac{C_2 w_g^2 T}{A^2} \quad \text{in. of water} \tag{82}$$

in which

D_5 = draft loss in straight ducts, in. of water.
D_5' = draft loss in duct bend, in. of water.
A = cross-sectional area of duct, sq. ft.
w_g = gas flowing, lb. per second.
P = perimeter of duct, ft.
L = length of duct, ft.
T = temperature of gases, °F., absolute.

Coefficients (at sea level):

C_1 = 0.0000803, circular steel duct.
= 0.0000187, brick or brick-lined duct.
C_2 = 0.0000803, 90-deg. sharp bend.
= 0.0000219, 135-deg. sharp bend.
= 0.0000183, 90-deg. bend (radius = duct diameter).
= $0.000073\left[\frac{A_2}{A_3} - 1\right]$, changes in duct area.
A_2 = cross-sectional area of large duct, sq. ft.
A_3 = cross-sectional area of small duct, sq. ft.

For calculating the draft loss through an economizer, the following equation is used:

$$D_4 = \frac{6.606}{10^{12}} F^2 LT \quad \text{in. of water} \tag{83}$$

in which

D_4 = draft loss, in. of water
F = gas flow per lineal foot of pipe in economizer section, lb. per hour per foot.
L = number of economizer sections.
T = average gas temperature, °F., absolute.

The loss through an *economizer* ranges from 0.2 to 1.0 in. and over. Air preheaters ordinarily produce a somewhat greater loss than do economizers, depending on the size, etc. In addition to those already mentioned are the losses due to dampers and sudden enlargements in the gas ducts. Ordinarily, these may be neglected.

The draft losses are necessarily cumulative. On adding the various increments mentioned, the *total draft* required at the chimney,

or the *available draft*, as it will be referred to later, will be obtained. This may be represented by the following equation:

$$D_a = D_1 + D_2 + D_3 + D_4 + D_5 \quad \text{in. water} \qquad (84)$$

in which D_a = the total available draft in inches of water, at the chimney entrance, and D_1, D_2, D_3, D_4, D_5, etc. = the various aforementioned increments of draft loss in inches of water.

If the draft loss due to velocity of flow and the chimney losses resulting from friction, leakage and cooling of the gases were added to D_a, of Eq. (84), the total, maximum, theoretical draft for the unit in question could be obtained. In practice, however, these losses are difficult to determine, and the available draft D_a is usually taken as 80 to 85 per cent of the theoretical.

Draft may be natural or artificial, depending on whether it is produced by chimneys or mechanical equipment, respectively.

123. Natural Draft.—Natural draft is produced by the use of stacks or chimneys. The terms "stack" or "chimney" may be used interchangeably though they are often used to distinguish between steel and either concrete or masonry construction, respectively.

The intensity of draft produced by this means is a measure of the unbalanced unit pressure resulting from the relative weights of the hot gases within and the cold air surrounding a chimney. Thus, a column of hot and low-density gas will exert less unit pressure at its base than will a column of cold and higher density air of equal height. Such a condition exists in all chimneys and it is a natural cause, producing a continuous flow of air and hot gases through the gas passages of a boiler unit. The unbalanced pressure must be sufficiently great to induce flow and overcome the various restrictions or *draft losses*.

Aside from producing draft, a chimney may have the function of disposing of smoke and waste gases high above the surrounding territory, to aid in avoiding their becoming a public nuisance. This often necessitates building chimneys much higher than is required for draft alone.

Natural draft is limited chiefly to small-sized units. It may be used in combination with mechanical draft, but is superseded completely by mechanical draft in large central power stations. Chimneys, nevertheless, are a necessity for practically all steam plants.

124. Types of Chimneys.—Chimneys for power plants are of steel, masonry or concrete construction. The general features of construction for masonry and concrete chimneys are quite similar. Those built of steel may be guyed or self-sustained.

125. Guyed Steel Chimneys.—Guyed steel chimneys are found among the smaller sizes and may be designed to rest on a separate foundation or on the breeching or framework above the boiler. They are of light construction and are usually built without a lining. The height and diameter seldom exceed 100 ft. and 6 ft., respectively. Their chief advantages are low cost, ease of construction, and ability to resist infiltration of air. On the other hand, steel chimneys must be painted periodically to prevent corrosion, and, if unlined, they effect considerable cooling of the hot gases within. Figure 103 illustrates a common type of construction for a guyed steel chimney. Figure 104 shows a method of supporting the same type of chimney on the breeching of a boiler.

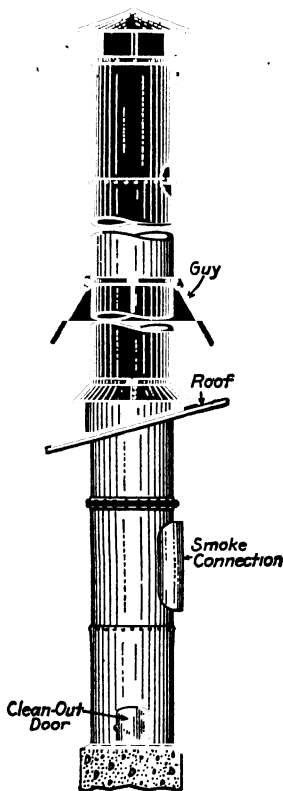



FIG. 103.—Guyed steel chimney.

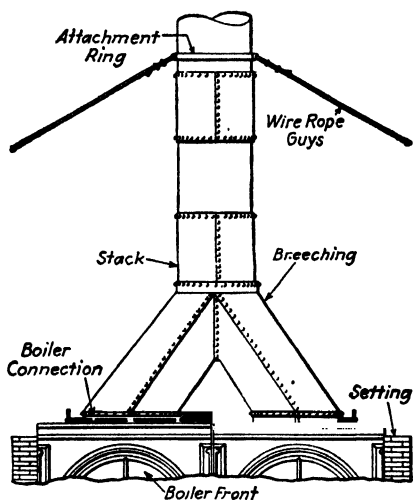



FIG. 104.—Showing the method of supporting a guyed steel chimney on the breeching of a boiler.

126. Self-sustaining Steel Chimneys.—Chimneys of this type are built in much larger sizes than are those of the guyed type. They may be built on a foundation or supported on steel work above the boilers. In either case the support is wholly dependent on the fastenings at the base. This type of chimney is most satisfactory when built with a lining, which may or may not be independent of the steel shell.

Independent linings are usually composed of suitable brick, set close, and grouted to the shell. The top portion is made approxi-

mately the width of one brick in thickness. The thickness increases in steps toward the bottom, thus providing a suitable foundation for the upper courses.

Supported linings are usually of especially prepared insulating materials, and rest, in sections, on projections or angle-iron shelf rings riveted to the shell. It is not necessary to employ tapered construction for this type of lining, except, perhaps, in a case where the chimney is to handle extremely hot gases, and where more insulation is required near the entrance.

A self-sustaining steel chimney should be provided with a ladder. Very often they are built with crown caps, especially when a lining is used. Figure 105 illustrates a self-sustaining steel chimney having these features. The bottom course is flared to give greater strength and to facilitate fastening to the foundation. In general, the first cost of a steel chimney of this type is less than for one built of concrete, brick or tile. They are, however, shorter lived and involve additional maintenance cost.

127. Brick Chimneys.—Chimneys built of *brick* are of a type by far most commonly used for steam-power plants. Common brick or perforated radial brick may be used in their construction. The chimney structure is generally supported on a concrete foundation, suitably built to prevent settling or tipping due to the action of the wind.

128. Common-brick Chimneys.—Chimneys built of common brick are usually round, though square and octagonal cross-sections are not uncommon in the smaller sizes. A lining extending some distance above the flue opening or all the way to the top is provided. This gives a type of construction which avoids cracking due to unequal expansion of the inner and outer surfaces. The chimney is usually tapered, and both the lining and the outer shell are built in sections, each section toward the top being reduced in thickness. The steps thus provided are on the adjacent surfaces, giving one smooth surface in contact

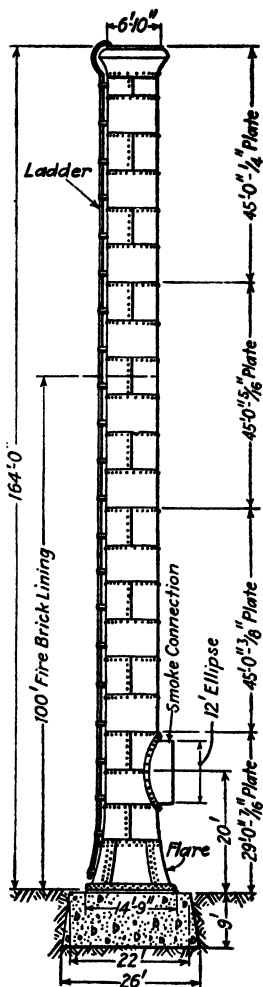



FIG. 105.—Self-sustaining steel chimney.

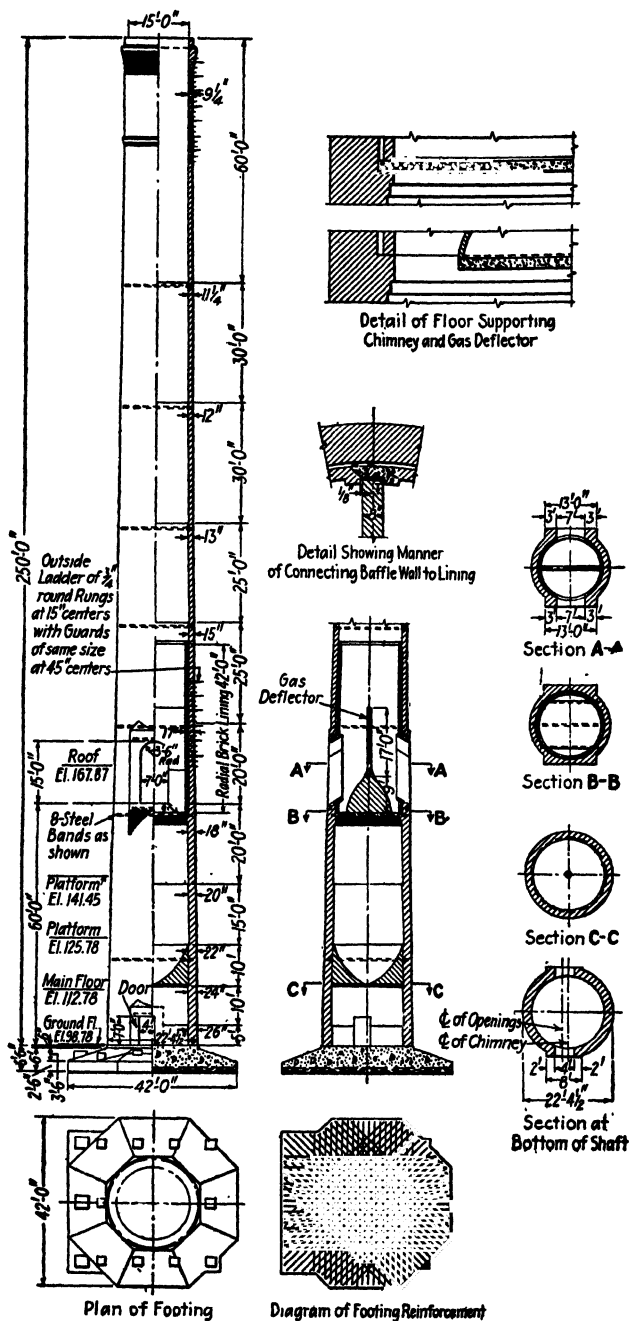



FIG. 106.—Radial-brick chimney, Purdue University power plant.

with the gases and another on the outside of the chimney. The air space between the lining and outside wall is at least 2 in. wide and aids slightly in reducing the loss due to radiation. Unless skill and selected materials are used in the construction of common-brick chimneys, excessive air leakage through the walls results. Because of this, common brick, for chimney construction, is rapidly being superseded by *perforated radial brick.*

129. Radial-brick Chimneys.—Chimneys built of this material are neat appearing, strong, and the draft loss due to radiation and air leakage through the walls is slight. Except for having thinner walls,

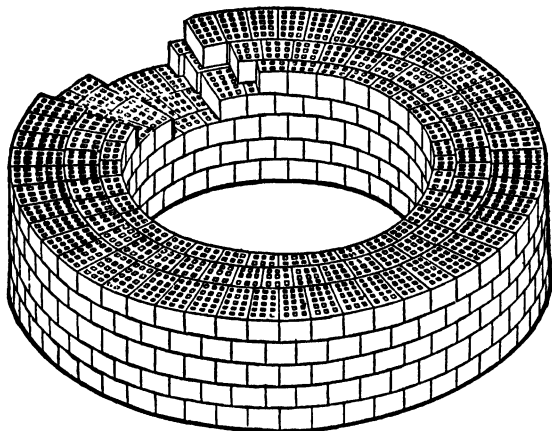



FIG. 107.—Showing method of laying perforated radial brick.

the construction differs very little from that where common brick is used.

Figure 106 illustrates a typical radial-brick chimney. They are built in all sizes, with full and short linings. The brick forming the outer walls are of varied lengths, and are curved to suit the radius of the chimney. The method of laying is illustrated in Fig. 107. The perforations seldom exceed 1 in. square (6 perforations in the smallest brick and 15 in the largest), and they aid in securing a good grade of brick during burning and provide an excellent bond for the mortar in the horizontal joints. Radial-brick chimneys for other than power purposes have been built as large as 60 ft. in diameter at the top and nearly 600 ft. high.

130. Reinforced-concrete and Tile Chimneys.—Chimneys built of *reinforced concrete* are coming into greater favor. Though unusual skill is required in their construction, if properly built, a strong monolithic structure having many distinct advantages results. They occupy less space, can be built rapidly, are generally tighter and cost

FIG. 108.—Weber reinforced-concrete chimney.

less than chimneys built of brick. The walls are comparatively thin, and because of their light weight only a small foundation is required. Figure 108 illustrates, in general, the type of construction used in reinforced concrete chimneys.

Tile in combination with reinforced concrete is sometimes used in the construction of chimneys. The tile have radial surfaces to suit the curvature of the walls, and, when in place, they are keyed together. Reinforcing bars are passed through the hollow portion of the tile and concrete poured around them, giving a solid structure with a smooth inner and outer surface.

131. Fixtures and Maintenance of Masonry and Concrete Chimneys.—Masonry and concrete chimneys should be provided with suitable lightning rods and a ladder extending to the top. A clean-out door at the base is also necessary. No painting and practically no maintenance is required on chimneys of this type. As a protection against the weather, the top of a masonry chimney should be provided with a suitable tile, concrete or cast-iron cap.

132. Conditions Which Determine Chimney Size.—The height of a chimney necessary to produce a required amount of draft is dependent chiefly on the draft losses through the unit and its altitude above sea level. The area required depends on the volume and rate of flow of the gases handled. However, the formulas used in calculation of chimney dimensions are largely of an empirical nature. The large number of variables which lend influence make a detailed analysis quite impractical.

133. Height of Chimney.—For determining the height, a tight system may be considered, at operating temperature and having all openings to the boiler furnace closed. In such a case there would be no flow of gases, and the draft below the grate (or in the burner box) would be purely static. The intensity of the draft measured under these conditions would be determined by the pressure difference between the hot gases within the system and the atmosphere without. If a draft gage is attached, say at a point below the grate, the fluid in the gage will rise until an equilibrium is established. From this, we may write the following equation:

$$Hd_g + \frac{Dd_w}{12} = Hd_a$$

$$D = \frac{12H}{d_w}(d_a - d_g) \qquad \text{in. of water} \tag{85}$$

expressing d_a and d_g in terms of the absolute temperatures of the air and gases by using $PV = wRt$. The volume V is taken as 1 cu. ft.,

and R is assumed to have the value 53.35 for both air and chimney gases.

$$d_a = \frac{P_b \times 144 \times 0.491}{53.35 \times T_a} = \frac{1.325P_b}{T_a}$$

$$d_g = \frac{1.325P_b}{T_g}$$

If d_w is taken as 62.3, for water at 70°F.,

$$D = \frac{12H}{62.3}\left(\frac{1.325P_b}{T_a} - \frac{1.325P_b}{T_g}\right)$$

$$= 0.255H\left(\frac{1}{T_a} - \frac{1}{T_g}\right)P_b \qquad \text{in. of water} \qquad (86)$$

If the available draft is taken as 80 per cent of the theoretical draft,

$$D_a = 0.204H\left(\frac{1}{T_a} - \frac{1}{T_g}\right)P_b \qquad \text{in. of water} \qquad (87)$$

In the above equations

D = theoretical draft, in. of water

D_a = available draft, in. of water

H = height of chimney above grate, ft.

T_a = temperature of air surrounding chimney, °F., absolute

T_g = average temperature of chimney gases, °F., absolute [the average temperature of chimney gases may be taken as 0.8 of temperature entering the chimney, average $t_g = 0.8t_g$ (entering)].

P_b = barometric pressure, in. of Hg.

The draft as calculated from Eq. (86) is theoretical static draft and could never be measured in actual practice. In case all doors and openings of a boiler unit connected to a chimney were suddenly closed, a pressure reading taken from below the grate would closely approximate the theoretical value. The following example illustrates the use of Eq. (86) in calculating the natural theoretical draft for an existing installation.

Example 7-1.—A chimney 200 ft. high is 120 in. in diameter. The gases, in passing through the chimney, have an average temperature of 500°F. The outside air temperature is 60°F., and the average barometric pressure is 28.5 in. of mercury. Calculate the maximum theoretical draft for the given conditions.

Solution.—Using Eq. (86) and substituting the values, $P_b = 28.5$, $H = 200$, $T_a = 460 + 60 = 520$, and $T_g = 460 + 500 = 960$, the draft, D, may be calculated. Thus,

$$D = 0.255 \times 200(1/520 - 1/960) \times 28.5$$
$$= 1.28 \text{ in. of water}$$

The following example illustrates the use of Eq. (87):

Example 7-2.—Required, the height of a chimney to serve a steam-generating unit consisting of 4 longitudinal-drum boilers (2 on each side of chimney), chain-grate stokers, and 2 breechings. The draft losses are as follows:

Through the grates	0.328 in. of water
Through the boiler	0.547
Due to bends in uptakes	0.102
Due to bends in breeching	0.102
Due to friction in breeching	0.028
Total available draft required	1.107 in. of water

Temperature and barometric pressure conditions are as in Example 7-1.

Solution.—Substituting the values, D_a = 1.107 in. of water, P_b = 28.5 in. of mercury, T_a = 520°F. and T_g = 960°F. in Eq. (87) and solving,

$$H = \frac{1.107}{0.204 \times 28.5(1/520 - 1/960)}$$
$$= 216.9 \text{ ft.}$$

134. Chimney Area and Diameter.—For determining the required cross-sectional chimney area for given set of conditions, the velocity of the gases may be assumed to be between 15 and 35 ft. per second. Knowing the combustion rate, and having calculated the total volume of waste gases, the area may be readily determined. Thus,

$$A = \frac{Q}{V_g} \tag{88}$$

in which

A = chimney area, sq. ft.
Q = rate of flow of waste gases, cu. ft. per second.
V_g = velocity of flow, ft. per second.

Having calculated the area, the diameter of a chimney is readily obtainable.

Example 7-3.—The chimney in Example 7-2 is to handle the gases from the boilers when 9,000 lb. of coal per hour are being burned. Assuming 18 lb. of waste gas per pound of coal and a maximum gas velocity of 30 ft. per second, calculate the required diameter for the chimney. Temperatures and barometric pressure are as given in Example 7-1.

Solution.—Using $PV = wRT$,

$$V = (1 \times 53.35 \times 960) \div (14.0 \times 144) = 25.4 \text{ cu. ft. of gas per pound.}$$

where

$P = P_b \times 0.491$ = 14.0 lb. per square inch barometric pressure.
w = 1 lb.
T = 500 + 460 = 960°F. (absolute temperature of gases).
R = 53.35 (constant as for air).

From Eq. (88)

$$A = \frac{Q}{V_g} = \frac{9{,}000 \times 18 \times 25.4}{30 \times 60 \times 60}$$
$$= 38.1 \text{ sq. ft.}$$
$$\text{Diameter} = 12\sqrt{\frac{4A}{\pi}}$$
$$= 84 \text{ in.}$$

The chimney dimensions, as calculated from the foregoing equations, are suitable for average conditions and are generally varied one way or another to meet other influencing factors not entering into the calculations.

135. Kent's Chimney Formula.—Kent's formula is often used in the calculation of chimney dimensions. It is based on data obtained from actual installations, and assumes a dead-air space of 2 in. around the inner circumference of the chimney and a combustion rate of 5 lb. of fuel per hour per rated boiler horsepower Kent's formula is as follows:

$$\text{Boiler hp.} = 3.33(A - 0.6\sqrt{A})\sqrt{H} \qquad (89)$$

in which

Boiler hp. = rated boiler horsepower.
A = actual area of chimney, sq. ft.
H = actual chimney height, ft.

136. Advantages of Mechanical Draft.—The use of mechanical or artificial means for producing draft has many distinct advantages over the natural-draft system. Mechanical-draft systems, however, are justified only where an economic gain can be realized, and this is one of the chief reasons for their widespread adoption for medium- and large-sized steam-power units. In small plants mechanical draft is often desirable, but in large central stations mechanical-draft systems are indispensable. The chief advantages may be summarized as follows:

1. Increased boiler capacity.
2. Saving in operating costs.
3. Higher efficiency.
4. Positive and flexible control.

A comparison of the first cost plus the full maintenance and operating expense of mechanical- and natural-draft systems of equal capacity should necessarily be made before one or the other is adopted. The choice, however, is nearly always dictated by experience, which generally shows mechanical draft to be the cheaper when more than 1.5 in. of water draft is required, and where the equipment is to be

operated at high ratings. The use of cheaper fuels and the greater *efficiency, attainable with mechanical draft, aid in effecting a lower cost of operation. The improved efficiency results; mainly, from the* advantages in control. Mechanical-draft equipment is readily adaptable to automatic regulation, which manifests its real value.

137. Types of Mechanical-draft Systems.—Artificial-draft systems for producing draft may be composed of steam jets, blowers or fans, and they are generally classified as

1. Forced draft.
2. Induced draft.
3. Balanced draft.

In the case of *forced draft*, the air for combustion is supplied to the point where it enters the furnace, at sufficient pressure to overcome the resistance offered by the grate and fuel bed or burner, as the case may be. *Induced draft* is produced by creating a sufficient vacuum in the gas passage on the discharge side of the boiler equipment to cause a flow, in much the same manner as with natural draft. *Balanced draft* is obtained by combining the forced- and induced-draft systems so as to maintain atmospheric pressure or a slight vacuum in the furnace.

138. Automatic Combustion Control.—The function of the combustion-control system is to supply fuel and air to the fuel-burning equipment in proper proportions and at a rate to permit the boiler to meet the demand for steam. Maximum combustion efficiency is obtained by the use of the minimum excess air required for complete combustion of the fuel. This condition results in the maximum temperature of the gaseous combustion products, which produces the greatest heat transfer to water and steam. In refractory furnaces it may prove undesirable to attain this condition of maximum boiler efficiency. The high temperature in the furnace may cause such an increase in refractory deterioration that the gain in efficiency is more than offset by the coincident increased maintenance expense. These factors must be balanced and the excess air must be increased for most economical results.

In proper control the following factors must *be* regulated: first, control of fuel and air rate to maintain constant steam pressure; second, control of air-fuel ratio for most economical combustion; third, control of draft in furnace, this being an indication of proper balance between induced and forced draft. Maintaining a constant steam pressure indicates a perfect balance between boiler B.t.u. input in fuel and the B.t.u. required by the prime mover.

In Fig. 109, combustion control is effected by regulating the speed of the drive of the following apparatus: stoker, forced-draft fan, and

induced-draft fan. The air or gas flow is often regulated by means of dampers. Combustion control may be entirely automatic, a combination of automatic and manual operation, or wholly manual. In any case, indicating instruments show operating conditions.

There are many types of automatic combustion-control equipment. Only one make will be described in this section, and the type selected

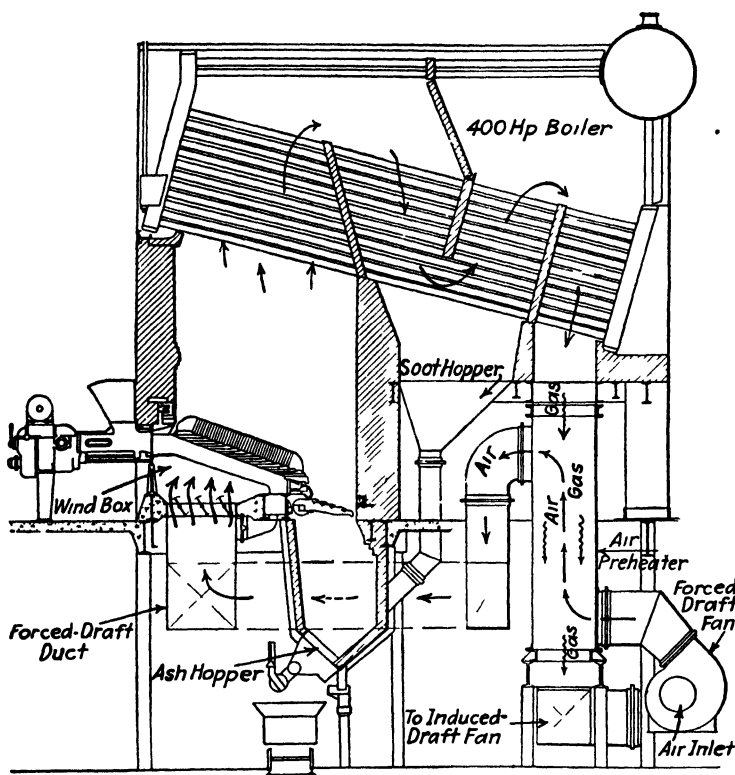



FIG. 109.—Typical forced-draft installation for an underfeed stoker.

is the air-operated control of the Bailey Meter Company. This is designed to meet the needs of the industrial plant and smaller central station. It is simple, effective, flexible and comparatively inexpensive. This system has three functions: control of fuel and air by steam pressure; readjustment of the fuel-air ratio by the steam-flow, air-flow meter; and the maintenance of balance between forced and induced draft by furnace pressure.

A diagrammatic layout of the air-operated control system as applied to a stoker-fired boiler is shown by Fig. 110. The master

pressure controller (Fig. 111), of which there is one for the entire plant, responds to pressure changes in the steam header and controls all boilers in the plant through the master air-pilot valve. The controller

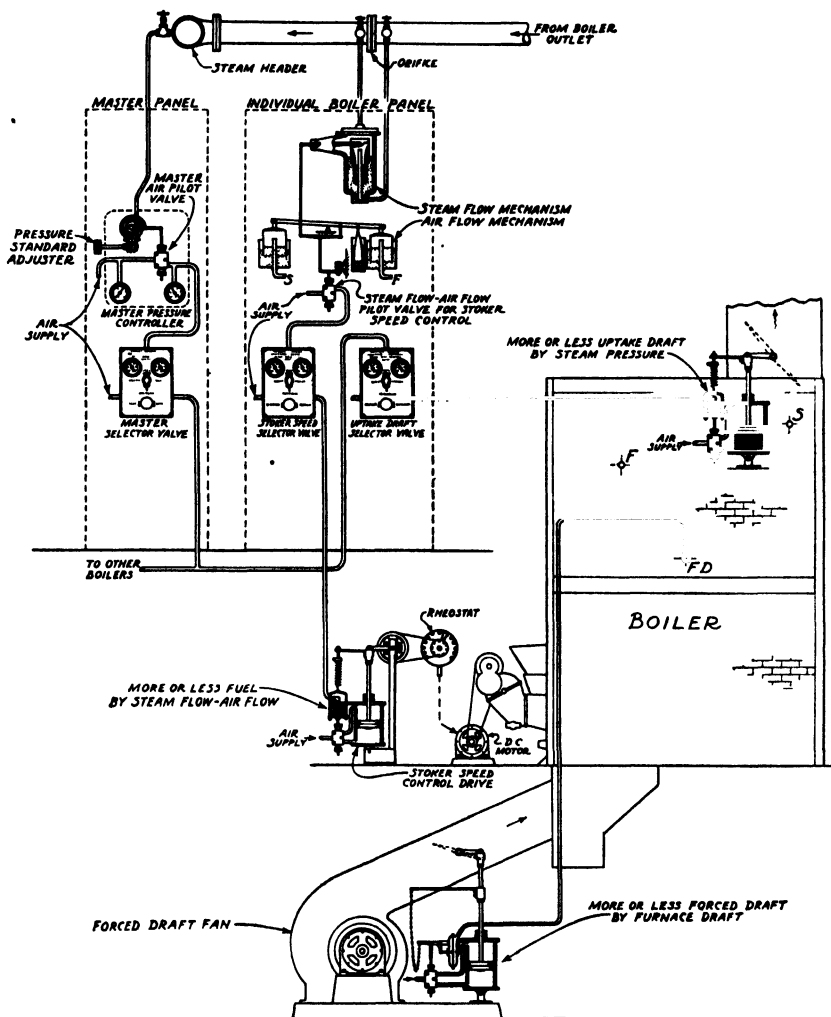



FIG. 110.—Air-operated combustion control. (*Bailey Meter Company.*)

is of the Bourdon helix type with an adjusting sector by which the pressure of all boilers may be raised or lowered as desired.

In practice, the controller (Fig. 111) is usually made of the recorder type, mounted in a case with two small pressure gages, one showing the air-supply pressure, the other the air pressure in the control system commonly referred to as the loading pressure.

In operation the system works as follows: An increase in load would cause a proportional drop in steam pressure which actuates

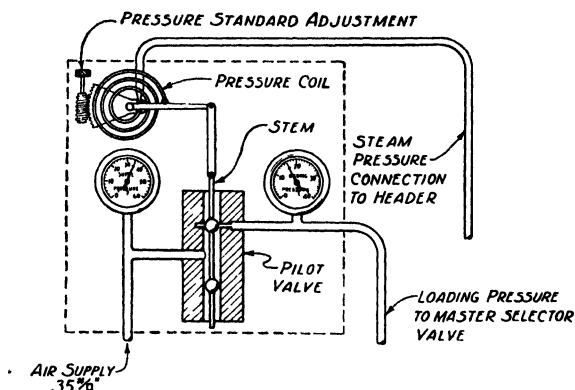



FIG. 111.—Controller (pressure). (*Bailey Meter Company.*)

the master controller to reposition the master air-pilot valve and increase the loading pressure. This increase in pressure is directed through the master selector valve (see Fig. 110) and the uptake draft selector valve (on each of the individual boiler panels) to a metal bellows on the uptake draft-control drive. The increase in pressure expands the bellows and displaces the relay-pilot valve (Fig. 112) so as to admit air pressure to the operating piston and open the damper an amount proportional to the change in control pressure. The pilot valve is returned to neutral position by means of the spring and yoke shown in Fig. 113, the tension of the spring being increased as the piston rod moves upward.

FIG. 112.—Bailey free-floating pilot valve.

As the damper opens, the furnace pressure tends to drop, but this change is immediately detected by the diaphragm-type furnace-draft regulator (Fig. 114) which moves to open the forced-draft damper and maintain the furnace draft constant. This increased flow of air through the boiler results in an increased differential across the points

F-S (see Fig. 110), and this differential connected to the air-flow side of the boiler meter is used as a measure of the air flow.

The pilot valve for the stoker-speed control is actuated by both the steam flow and the air flow. The linkage is designed so that the pilot valve will have, for every value of steam flow (when the pens, air flow and steam flow, are together) a definite position which determines the loading pressure sent to the stoker-control drive. If the pens are not together the pilot valve will be displaced from its normal position in a direction to bring them together by changing the stoker speed. Thus if the air flow rises above the steam flow, indicating excess air, the stoker will be speeded up so as to maintain the desired ratio.

Fig. 113.—Bailey air-operated control drive.

Stoker speed is regulated by a drive unit similar to that used on the uptake damper. The furnace draft is maintained by an oil-sealed bell or large sensitive diaphragm drive. In the diaphragm type, as in Figs. 110 and 114, linkage from the diaphragm moves the pilot valve to control the movement of the piston. Motion of the piston repositions the pilot by means of a chain, diminishing or adding to the counterbalance weight until the pilot valve is again in its neutral position. If it is desired to have the furnace draft subject to either manual or automatic control, an oil-sealed bell and standard control drive are utilized and a selector valve is added to the individual boiler panel.

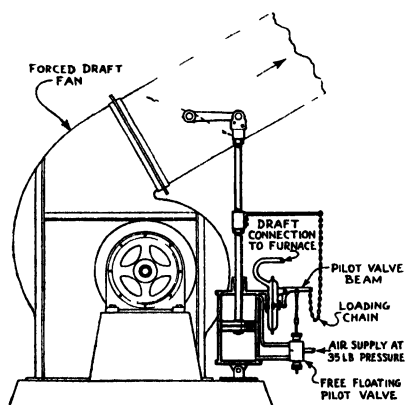



Fig. 114.—Diagrammatic drawing of Bailey furnace-draft controller.

The selector valves provide the flexibility of the system, allowing either manual or automatic control and collective or individual adjustment. One selector valve is provided for the master panel. With the transfer knob in the manual-operation position, the output of all boilers may be governed manually using the right-hand pressure gage as an index. Variation of this gage from 5 to 25 lb., as effected by

turning the adjustment knob, represents a variation in plant output from minimum to maximum. Individual boiler panels may also be furnished with selector valves for those factors for which flexibility of operation is desired, for example, stoker speed and uptake draft. In this manner, each of these factors can be taken off automatic control and operated manually if desired.

On any of the selector valves, with the transfer knob in either manual or automatic position, the right-hand pressure gage is an index of the speed or position of the controlled factor. The left-hand gage indicates the loading pressure received at the selector valve from the controller.

The air-pilot valve shown by Fig. 112 is the heart of the system and is used in practically every individual piece of equipment making up the control. It consists of a balanced, frictionless, free-floating stem with two balls of equal size controlling the air flow through ported sleeves. Air is admitted at constant pressure to the space between the two balls, which, being slightly smaller in diameter than the pilot valve sleeves, allow a small quantity of air to escape. This tends to prevent actual contact between the balls and the sleeves and acts as a lubricant.

The pilot valve shown is of the double-acting type used as a relay with the piston-operated control drive. The two outlets are connected to the control-drive cylinder. When the pilot valve is in the neutral position as illustrated, the air pressure supplied to each side of the piston produces equal opposing forces and the piston remains stationary. Should the pilot be moved slightly in either direction, increased air pressure would be admitted through one port to one side of the piston and air would be exhausted from the opposite side through the other port.

To obtain a single-acting pilot valve such as required for use with the master and steam-flow, air-flow controllers, one outlet is plugged. Each position of the pilot-valve stem will result in a definite pressure being set up in the outlet port varying between 5 and 25 lb. per square inch for full-range movement of the pilot.

Clean compressed air at 35 lb. per square inch pressure is required for supplying the controller pilot valves and the control-drive pilot valves. The source should be of sufficiently high pressure so that this value can be maintained at all times by pressure-reducing valves.

A main air filter and pressure-reducing valves are installed in the air-supply lines to insure having clean dry air at constant pressure. Individual filters are also provided in the lines to and from each pilot so that any dirt which may be in the connecting lines will be prevented

from interfering with the operation of the valve. The volume of air required is relatively small and may be figured approximately at 0.5 cu. ft. per minute for each pilot valve and selector valve in the system.

FIG. 115.—Prat-Daniel system of draft.

139. Mechanical-draft Equipment.—Steam-jet or blower equipment for the production of draft is sometimes used with hand-fired grates. In locomotive practice the exhaust steam from the cylinders is discharged through a jet below the stack, by this means inducing the necessary draft. Equipment of this sort is designed and arranged to effect the desired action by means of the principle involved in the operation of aspirators and ejectors. Figure 116 illustrates the application of an undergrate steam blower to a hand-fired grate. Steam and air are discharged into the ash pit through the Venturi-shaped nozzle. The steam jets are arranged in an annular manifold suspended in the nozzle, as shown. Jets and blowers may also be installed in small chimneys, and they are often used in this way for emergency operation.

These methods of creating draft are generally uneconomical to operate unless the steam used would otherwise be wasted.

Small *propeller-type blowers* are sometimes used with hand-fired

FIG. 116.—McClave undergrate blower.

grates for obtaining a forced draft. This type of fan is either motor or

turbine driven and is installed in much the same manner as steam blowers for this purpose. Figure 117 shows a section through a propeller-type blower driven by an impulse steam turbine. The exhaust steam may be used for heating feedwater.

The Prat-Daniel draft system (Fig. 115) uses both forced-draft and induced-draft fans with Venturi-shaped chimneys. The induced-draft fan discharges a jet of outside air (outer-circuit type), or a jet of flue gases (inner-circuit type), through a nozzle at the base of the

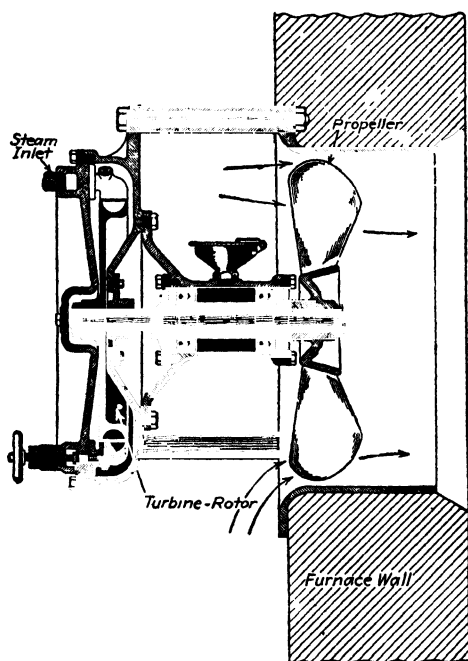



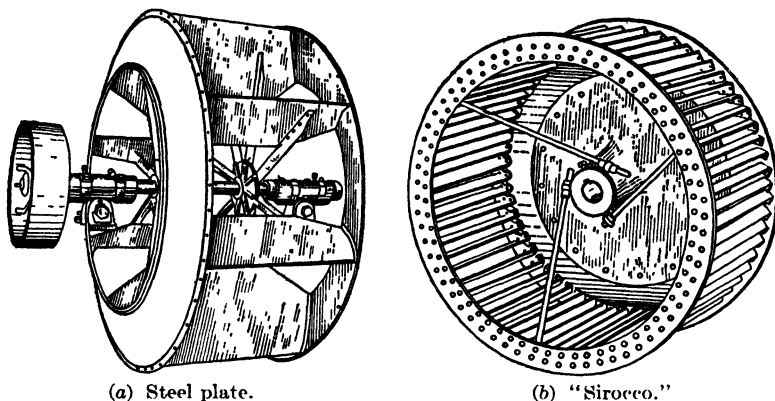
FIG. 117.—Sturtevant turbo-undergrate propeller blower.

"diffuser." This creates a partial vacuum and induces a flow of gas through the boiler passes.

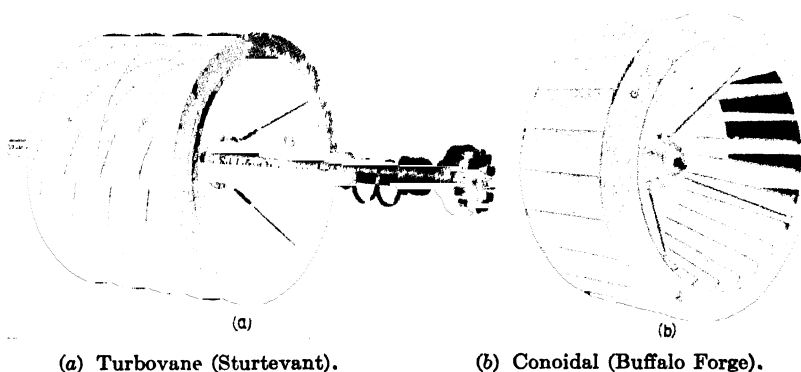
Fans most commonly used for forced-draft and induced-draft service have cylindrical impeller wheels running in an involute or scroll housing. The blade tips form the elements of a cylinder, and the blades extend some distance toward the axis. The shape of the blades determines the fan characteristics. They may be straight, or forward or backward curved with respect to the direction of rotation. Combinations of these shapes are also used. Fans of this type might be called "cylindrical" though in practice they are generally classified as follows:

1. Steel plate.
2. Multiblade.

The impeller blades and fan housings are made from sheet steel.

Steel-plate fans have impeller wheels containing from 5 to 12 straight or slightly curved blades fastened on spider arms, as shown in Fig. 118 *a*. Air is taken into the impeller along the direction of its axis and discharged more or less radially into the housing. The flow from the

(*a*) Steel plate. (*b*) "Sirocco."
FIG. 118.—Multiblade fan impellers

housing is tangential, and the discharge opening may be arranged at any angle. Steel-plate fans are used for all types of service and generally cost less than multiblade fans.

In *multiblade* fans the impeller wheels have a large number of blades, and in practically all cases the blades are either single or double

(*a*) Turbovane (Sturtevant). (*b*) Conoidal (Buffalo Forge).
FIG. 119.—Multiblade fan impellers.

curved. Figure 118 *b* illustrates the Sirocco fan, a multiblade fan having short and forward curved blades. This fan is used extensively for induced-draft service, where large volumes of gas or air are to be

handled. Other types of multiblade impellers are shown in Fig. 119. The turbovane impeller is ruggedly built, with backward curved blades and is designed for high-speed service. The conoidal impeller contains triangular-shaped blades, each having a backward curve at the inlet and a forward curve on the discharge.

Fans of both classes are built with inlets on one or both sides of the housing, in which cases they are termed "single" or "double inlet," respectively. Figure 120 shows an assembled fan of the double-inlet type. The terms "full housed," "7/8 housed" and "3/4 housed" are used in fan practice to indicate the portion of the housing above the supporting base.

FIG. 120.—Buffalo forge double-inlet fan.

Fans may be belt or direct connected to their drives, depending on the service. For low-speed fans, small steam engines are often used for driving, while direct-connected motors or turbines are required in cases where the speed is high. The flow of air may be controlled by damping or by varying the fan speed. Either method is susceptible to automatic control, but the latter is more desired from the standpoint of economy.

140. Fan Theory.—The following nomenclature is used by the Standard Test Code for Disc and Propeller Fans.

Centrifugal Fans and Blowers:

Standard air is air weighing 0.07488 lb. per cubic foot, corresponding to 29.92 in. Hg absolute pressure, dry-bulb temperature of 68°F., and 50 per cent relative humidity.

Static pressure, p_s, is measured at right angles to the direction of flow.

Total pressure, p_t, is measured by an impact tube.

Velocity pressure, p_{av}, is the difference between the total pressure and the static pressure.

Capacity, Q, is the cubic feet of air per minute handled by the fan.

Horsepower output, a.hp., is the air horsepower of the fan.

Horsepower input, hp. or brake hp., is the horsepower required to drive the fan.

Mechanical efficiency, e_m, is the ratio of the horsepower output to the horsepower input.

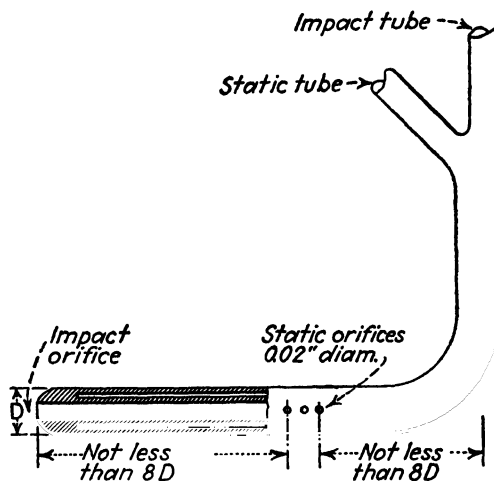



FIG. 121.—Pitot tube.

Static efficiency, e_s, is the mechanical efficiency multiplied by the ratio of static to total pressure.

Fan performance is a statement of the capacity, pressure or pressures, speed, and horsepower input.

Fan characteristic is a graphical presentation of fan performances throughout the full range from free delivery to no delivery at constant speed.

For measuring pressures the double Pitot tube (Fig. 121) is used. This is arranged to measure both static pressure, by means of static orifices, and total pressure by the impact orifice.

There are not less than four static orifices, not exceeding 0.02 in. in diameter. For the number of readings necessary in the different ducts (round and rectangular), reference should be made to the Standard Test Code.

To understand the meaning of these different pressures, if the fan discharge is completely closed, only potential energy is present. The

air pressure present, called "static pressure" tends to burst the housing. The opposite case is where the air flow is entirely unrestricted and all of the energy results in producing flow. In this case the air possesses kinetic energy only, and the pressure exerted by virtue of this energy is called "velocity pressure."

The measurement of the total, static and velocity pressures existing in ducts through which air or gas is flowing is illustrated in Fig. 122. The U-shaped tubes represent ordinary water manometers or draft gages, and the various pressures, p_s, p_t, and p_v, as shown, indicate the relationship between static, total and velocity pressure, respectively.

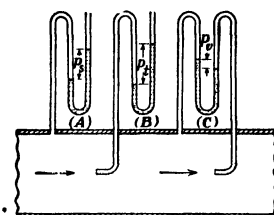



Fig. 122.—Illustrating relations between static, total, and velocity pressure.

In testing an exhausting fan an inlet duct is provided having a cross-sectional area equal to the fan inlet, and a length of at least 6 diameters (inlet duct). The total head or total pressure produced by the fan is the difference between the average *absolute total pressure* in the discharge duct and the average *absolute total pressure* in the inlet duct with a correction (addition) for the friction loss in the inlet and outlet ducts between the points of pressure measurement.

The average *absolute total pressure* in the inlet duct is determined by adding to the *absolute static pressure* in the inlet duct the calculated average *velocity pressure* in the inlet duct. This addition is algebraic if, instead of calculated absolute pressures, observed readings are used, those below atmospheric pressure being negative, those above atmospheric pressure positive, and the velocity head always positive.

Any pressures read on an impact tube are total pressures, and on a static tube are static pressures, whether the pressures read are above or below atmospheric pressure. They, however, should be expressed as absolute pressures, or must be assigned a positive or negative value, for pressures above or below atmospheric pressure respectively.

If a fan test is assumed with pressure readings taken both in the inlet and discharge ducts, in the inlet duct the impact-tube reading is 1 in. and the static-tube reading is 2 in., both below atmosphere, and in the discharge duct, of same size as the inlet duct, the impact-tube reading is 2 in. and the static-tube reading is 1 in., both positive. The total head against which the fan is operating is the difference between the readings on the discharge and inlet sides of the fan, with the readings taken with the same kind of tube.

With the impact tube,

$$p_t = 2 \text{ in.} - (-1 \text{ in.}) = 3 \text{ in.}$$

With the static tube,

$$p_t = 1 \text{ in.} - (-2 \text{ in.}) = 3 \text{ in.}$$

For methods of calculating friction, the number of readings required, and the effect of different shapes of ducts, reference should be made to the Fan Test Code.

In a fan with only a discharge duct delivering air at a pressure above atmospheric pressure the total pressure is determined as follows:

$$p_t = p_s + p_{av} \qquad \text{in. of water} \tag{90}$$

in which

p_t = total pressure, in. of water.
p_s = static pressure, in. of water.
p_{av} = average velocity pressure, in. of water.

Pressure measurements at right angles to the directions of flow may be obtained with a piezometer ring, a hollow annular ring having a series of small tubes connecting it to the duct wall, around the circumference. The piezometer ring is connected to the draft gage by means of rubber tubing. This arrangement facilitates getting an average pressure from the whole circumference of the duct.

For obtaining the velocity pressure direct, a commercial type of Pitot tube is often used. With this the duct may be divided into sections and a measurement taken in each. The readings thus taken should be averaged algebraically and the average value used for calculating purposes.

Because of the friction against the duct walls, the velocity of air at the center of the duct is maximum through the cross-section. The ratio between velocity at the center and the average velocity is given by the center coefficient, as follows:

$$C = \frac{V_a}{V_c} = 0.92 \qquad \text{(approximate)}$$

The well-known equation for flow, given below, is the basic equation used in calculating the *velocity of the fluid flowing*.

$$V = \sqrt{2gh}$$

in which

V = velocity, ft. per second.
g = acceleration of gravity, 32.2.
h = head in feet of fluid producing the flow.

A more usable equation may be obtained by substituting the value for g and expressing h in terms of gas density and average velocity pressure in inches of water. Thus,

$$V_a = 60\sqrt{\frac{2 \times 32.2 \times 62.3 \times p_{av}}{12 \times d}}$$

$$= 1{,}096\sqrt{\frac{p_{av}}{d}} \tag{91}$$

in which

V_a = average velocity, ft. per minute.
p_{av} = the average velocity pressure, in. of water.
d = density of fluid flowing, lb. per cubic foot.
62.3 = density of water at 70°F. in draft gage, lb. per cubic foot.

Having determined the average velocity, the *quantity of fluid flowing* may be calculated as follows:

$$Q = AV_a \tag{92}$$

in which

Q = quantity of flow, cu. ft. per minute.
A = cross-section area of duct, sq. ft.
V_a = average velocity of flow, from Eq. (91).

141. Fan Air Horsepower.—Fan air horsepower may be determined by use of the horsepower equation. Thus

$$\text{a.hp.} = \frac{62.3 \times Q \times p_t}{33{,}000 \times 12} \tag{93}$$

$$= \frac{Qp_t}{6356}$$

in which

a.hp. = fan air horsepower.
Q = quantity of flow, cu. ft. per minute.
p_t = total fluid pressure developed by the fan, in. of water.
62.3 = density of water at 70°F. in draft gage, lb. per cubic foot.

142. Fan Mechanical Efficiency.—The mechanical efficiency of a fan is a ratio of the air horsepower to the brake horsepower or horsepower input. Thus,

$$e_m = \frac{\text{a.hp.} \times 100}{\text{b.hp.}} \tag{94}$$

in which

e_m = fan mechanical efficiency, per cent
a.hp. = fan air horsepower, from Eq. (93).
b.hp. = brake horsepower or horsepower required at fan coupling or driving pulley.

Example 7-4.—A fan is required to deliver 18,000 cu. ft. of air per minute, at 70°F. and 3 in. of water, static pressure, to a boiler furnace. The duct is short and has an area of 9 sq. ft., and the mechanical efficiency of the fan at the capacity desired is 62 per cent. The density of the air delivered is 0.0749. Determine the air horsepower of the fan and the horsepower required to drive it.

Solution.—The average velocity in the duct is 18,000 ÷ 9 = 2,000 ft. per minute. Using Eq. (91),

$$p_{av} = 0.0749\left(\frac{2,000}{1,096}\right)^2 = 0.25 \text{ in. of water,}$$

and

$$p_t = 3 + 0.25 = 3.25 \text{ in. of water}$$

$$\text{a.hp.} = \frac{18,000 \times 3.25}{6,356} = 9.2$$

$$\text{b.hp.} = \frac{9.2}{0.62} = 14.84$$

143. Fan Performance.—Data of fan performance, either in tabular form or as characteristic curves, consist of *speed* (r.p.m.),

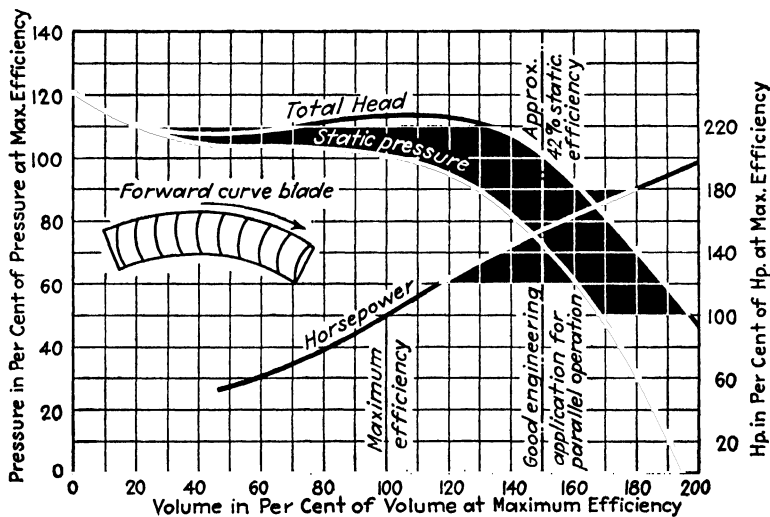



Fig. 123.—Characteristics, forward-curved fan.

horsepower, *capacity* (c.f.m.), *static pressure* and usually *total pressure* (inches of water). The *static efficiency* may be included. The curves shown indicate the performance of different types of fans for the range of capacity of each fan. The performance is plotted on a percentage basis, 100 per cent of wide-open volume representing maximum capacity. A fan is selected for service so that its normal operation will be at its most efficient point. This is generally at about 50 per cent of maximum capacity.

For forced draft, either for pulverized coal or stokers the backward-curved fan or the full-backward-curved fan fulfill the requirements of efficiency, pressure, and power. The full-backward-curved fan (Fig. 124) has high efficiency as a large part of the energy is static pressure when the air leaves the impeller, and only a comparatively small conversion from velocity to static pressure is required in the housing. If the fuel cakes or clinker forms, increasing the fuel-bed resistance, the air flow decreases. As seen by the steep pressure curve, lowering the capacity of the fan, even to a small degree, gives an

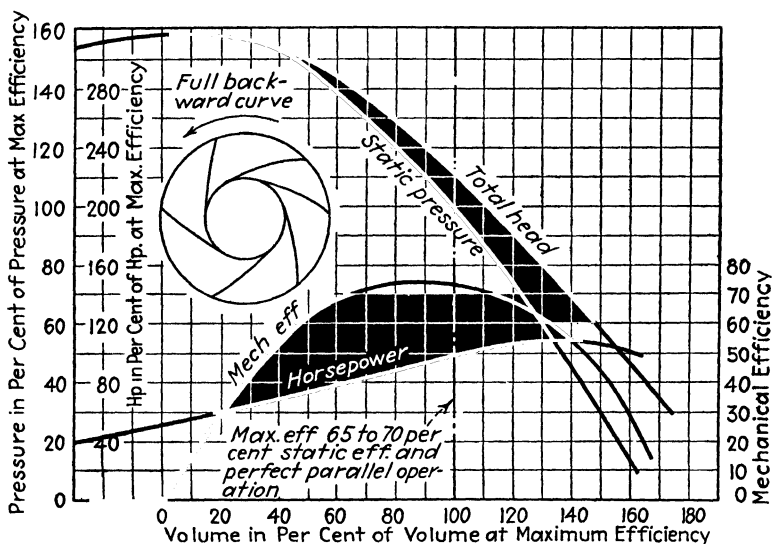



Fig. 124.—Characteristics, full backward-curved fan.

immediate large increase in pressure, opening up the fuel bed and maintaining the combustion rate.

Induced-draft requirements are different, and for this service the forward-curve (Fig. 123), steel plate or radial-tip fans are suitable. The induced-draft fan has to overcome the resistance to flow through the boiler, economizer, air preheater, and breechings. Because of the high temperatures, greater volumes are handled than in the case of the forced-draft fan. Gases contain cinders, ash, and soot which cause blade erosion and deposits on blades. The latter cause unbalanced stresses which increase with the square of the speed, therefore the fan is selected with the consideration of this advantage of low speed in preventing vibration. There is no need for pressure, but because of the high capacity required a fan with high efficiency over a wide range of capacity is selected.

The Buffalo Forge Company suggests that in practice for hand-fired boilers, 100 per cent excess air should be allowed, a total of 16.7 cu. ft. air per minute per boiler rated horsepower at 70°F. for a forced-draft fan, and 32.4 cu. ft. flue gases per minute at 550°F. for an induced-draft fan. For stoker-fired boilers, the allowance is for 50 per cent excess air, or 11.7 cu. ft. per minute per boiler rated horsepower, or 22.8 cu. ft. per minute at 550°F. Tables giving complete data are available.

Characteristic fan relations:

1. Capacity (c.f.m.) is directly proportional to speed (r.p.m.).
2. Head or pressure (inches of water) is directly proportional to $(\text{speed})^2$.
3. Horsepower is directly proportional to $(\text{speed})^3$.

AIR PREHEATERS

144. Purpose and Classification.—Air preheaters, sometimes called air economizers or simply air heaters, are used in steam boiler plants for the purpose of recovering a portion of the heat of the flue gases. This is accomplished by forcing the air for combustion over metal surfaces heated by the hot flue gases before discharging it into the furnace. The heat thus recovered aids in increasing the operating efficiency and capacity of the boiler unit.

Air preheaters may be divided into two types as follows:

1. Recuperative.
2. Regenerative.

145. Recuperative Air Preheaters.—This type of air preheater is constructed with plate or tube heating elements. Plate heaters consist of a number of thin sheet-like ducts contained within a suitable housing in such a way that the air may pass through the ducts and the gas over their outer surfaces. Figure 125 shows a phantom view of a plate air preheater. As shown in the figure, the air enters at the far side, above, and discharges below, from the near side. The flue gases pass through the preheater in a vertical direction only. The elements are spaced so as to allow sufficient opening for the passage of the flue gases.

Tubular heaters employ the same principle as those having plate elements. They contain a large number of small tubes extending between an upper and a lower tube sheet. A rectangular steel casing encloses the elements, and the arrangement is such that the gases pass through the tubes, and the air passes around them.

Recuperative air preheaters are generally designed on the counter-flow principle; that is so that the air enters in the region of the cooler

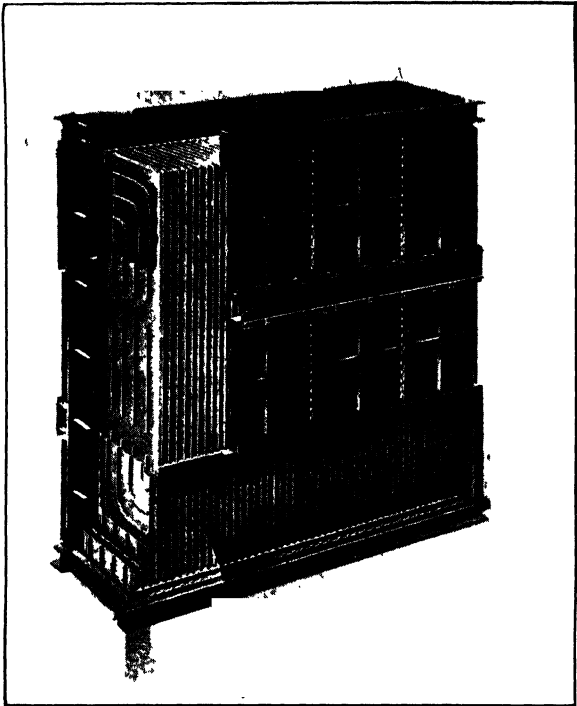



Fig. 125.—Phantom view, showing Combustion Engineering plate-type air preheater.

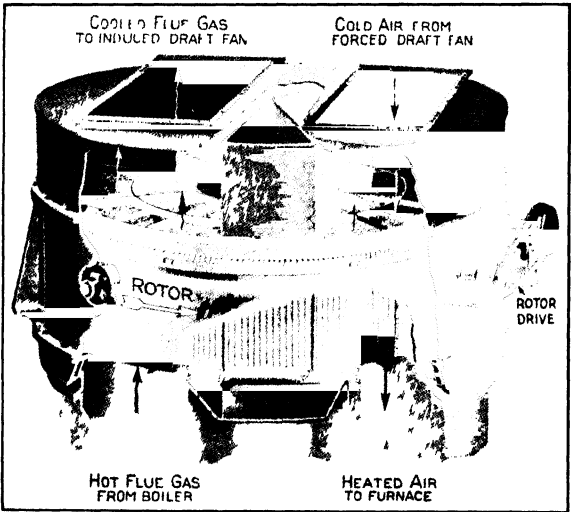



Fig. 126.—Ljungstrom air preheater.

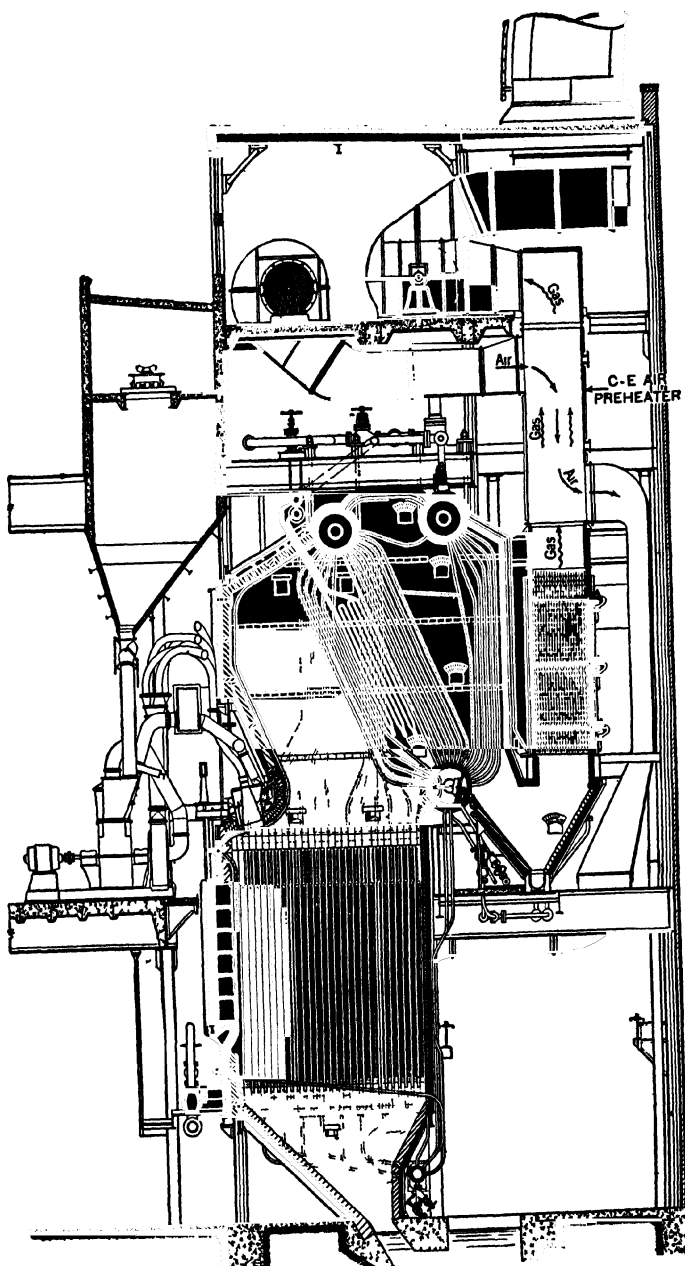



FIG 127 —Showing an installation of an air preheater with an economizer, 16,950 sq ft, 1,400 lb pressure, Ladd boiler, Northeast Station Kansas City Power & Light Co

gases and leaves in the region where the hot gases enter. They are built in all sizes and are widely used in power plants throughout the United States.

146. Regenerative Air Preheaters.—Air preheaters of this type are distinct from those of the recuperative type in that the heating elements move. The operation is such that heat is conveyed from the hot flue gases to the air by means of a honeycomb rotor. The Ljungstrom preheater (Fig. 126) is of this type. The rotor is of heavy steel construction and contains numerous passages for the air and gases. It absorbs heat from the flue gases in one section and, as it rotates, this heat is given up to the air in another section. Ljungstrom preheaters are built to operate in various positions and with air and gas connections from any direction. They are also built with forced- and induced-draft fans integral with the housing.

FIG. 128.—Theoretical compressor cycle without clearance.

In the new type of Ljungstrom heater the motor is placed on top of the frame driving the vertical shaft through bevel gears. In some installations the axis of the heater is horizontal.

147. Typical Installation.—At one time air preheaters were installed as an alternative to feedwater economizers. Present practice, however, aims at lowering the temperature of the flue gases as much as possible, and, to accomplish this, it often proves good economy to install both. Figure 127 is an illustration of such a case in a large utility plant. Higher combustion rates are thus permissible and greater efficiency results.

148. Theoretical Air Compressor.—The reciprocating-piston air compressor operates on a cycle which is the reverse of the ideal heat engine. Figure 128 illustrates the cycle of the ideal single-stage compressor, which, like the ideal heat engine, is assumed to have zero clearance. The work of compression is represented on the P-V diagram as the enclosed area $ABCD$, and depends upon the two pressures P_1 and P_2, the two volumes V_A and V_B, and the compression curve AB. For this development, the compression is assumed to follow the polytropic curve ($PV^n = C$), although the development would be similar for either adiabatic ($PV^k = C$) or isothermal ($PV = C$) compression.

The total work required for the cycle with polytropic compression may be calculated by the algebraic summation of the following areas:

$$W = \text{area } A_1ABB_1 + \text{area } B_1BCO + \text{area } ADAA_1$$

The areas A_1ABB_1 and B_1BCO will give negative results and area $ODAA_1$ positive in foot-pounds, the net work being negative indicating that work is performed on the air.

Substituting P, V values for the work areas:

$$W = \frac{P_A V_A - P_B V_B}{n - 1} + (-P_B V_B) + P_A V_A$$

$$W = \frac{n}{n - 1}(P_A V_A - P_B V_B) = \frac{n}{n - 1} WR(T_A - T_B)$$

since

$$\frac{T_A}{T_B} = \left(\frac{P_B}{P_A}\right)^{\frac{n-1}{n}} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

Substituting P_2 for P_B, and P_1 and V_1 for P_A and V_A, respectively,

$$W = \frac{n}{n - 1} P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right] \tag{95}$$

Equation (95) expresses the work of the ideal compressor cycle without clearance. The introduction of clearance, present in the

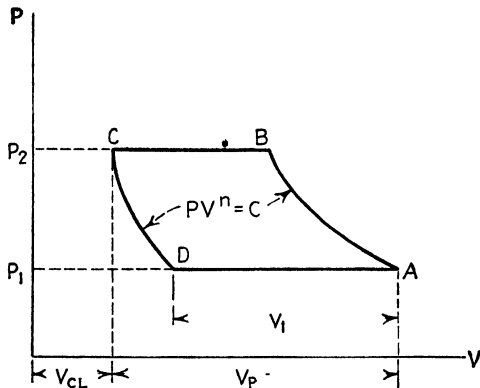



FIG. 129. -Theoretical air-compressor cycle with clearance.

actual compressor, will change the work of compression. The effect of clearance is to change the volume of air actually compressed each cycle as shown by Fig. 129, and by the following development. If V_P = piston displacement volume per cycle, cu. ft.:

$$V_{CL} = mV_P$$

in which

m = clearance, as a decimal part of V_P.

$$V_1 = V_A - V_D$$

$$V_D = V_C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} = V_{CL}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} = mV_P\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$V_A = V_P + V_{CL} = V_P + mV_P$$

$$V_1 = V_P + mV_P - mV_P\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$V_1 = V_P\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right] \qquad (96)$$

in which

V_1 = volume of air at intake pressure, actually compressed per cycle, cu. ft.

P_2 = discharge pressure, lb. per square inch absolute.

P_1 = intake pressure, lb. per square inch absolute.

This equation is based upon cubic feet per cycle. It is true also on the basis of a unit of time such as a minute. In this case, both V_1 and V_P are given as cubic feet per minute.

The volumetric efficiency is the ratio of the volume of air actually compressed to the volume of the piston displacement. Hence:

$$e_v = \frac{V_1}{V_P} = 1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \qquad (97)$$

Some authorities define volumetric efficiency as the ratio of the weight of air compressed per cycle in an ideal compressor with clearance to the weight compressed in an ideal compressor without clearance and for which $P_1 = P_{AT}$ (atmospheric pressure). Neglecting the heat flow between the cylinder and air, the following assumptions are made: $T_B = T_C$ and $T_D = T_A$. The volumetric efficiency is calculated as follows:

$$e_v = \frac{w_A - w_D}{V_P \times d_{AT}}$$

in which

d_{AT} = density of air at atmospheric pressure and temperature.

$$e_v = \frac{\frac{P_1}{RT_1}(V_A - V_D)}{V_P \times \frac{P_{AT}}{RT_1}} = \frac{P_1}{P_{AT}}\left[\frac{V_A - V_D}{V_P}\right]$$

$$e_v = \frac{P_1}{P_{AT}}\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right] \qquad (98)$$

Example 7-5.—Calculate the horsepower of an ideal single-stage double-acting compressor, 10 × 11 in., 2-in. piston rod, running at 300 r.p.m. Atmospheric pressure 14.7 lb. per square inch absolute, intake pressure 14.2 lb. per square inch absolute, discharge pressure 125 lb. per square inch absolute. The average clearance of the compressor is 4.5 per cent, and the value of n is 1.28.

Solution.—The volumetric efficiency of the compressor:

$$e_v = \frac{P_1}{P_{AT}}\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right] = \frac{14.2}{14.7}\left[1 + 0.045 - 0.045\left(\frac{125}{14.2}\right)^{\frac{1}{1.28}}\right]$$

$$= \frac{14.2}{14.7}[1 + 0.045 - 0.045 \times 5.47] = \frac{14.2}{14.7} \times 0.799 = 0.772$$

The piston displacement volume in cubic feet per minute:

$$V_P = \left[\frac{\pi \times 10^2}{4 \times 144} + \frac{\pi(10^2 - 2^2)}{4 \times 144}\right] \times \frac{11}{12} \times 300$$

$$= \frac{\pi}{4 \times 144}(100 + 96) \times \frac{11}{12} \times 300 = 294 \text{ cu. ft. per minute}$$

The volume of air at intake pressure compressed in cubic feet per minute:

$$V_1 = e_v V_P = 0.772 \times 294 = 227 \text{ cu. ft. per minute}$$

The compressor horsepower:

$$\text{hp.} = \frac{1.28 \times 14.2 \times 144 \times 227}{0.28 \times 33{,}000}\left[1 - \left(\frac{125}{14.2}\right)^{\frac{0.28}{1.28}}\right]$$

$$= \frac{1.28 \times 14.2 \times 144 \times 227}{0.28 \times 33{,}000}(1 - 1.61) = -39.2$$

The ideal compression would be the isothermal compression in which the heat generated in the air during the process is removed as produced. In this compression the work is a minimum and for the ideal compressor with zero clearance is:

$$W = P_1V_1 \log_e \frac{P_1}{P_2} \tag{99}$$

In practice, the cylinder walls are water-jacketed, but very little cooling occurs. In order to approach isothermal compression as near as practical, two-stage or multi-stage compression is used. In the two-stage ideal compressor, after partial compression in one cylinder, the air is discharged into an intercooler, in which the heat generated during the first compression is absorbed by cooling water. From the intercooler the air enters the high-pressure cylinder and is compressed to the discharge pressure. In the ideal two-stage compressor there is no pressure change through the intercooler, and the temperature of the air is the same entering the low-pressure and high-pressure cylinder.

The work of a two-stage compressor is obtained by the addition of the work of compression performed in the two air cylinders.

Referring to Fig. 130, the work in the first stage is

$$W_1 = \frac{n}{n-1} P_1 V_1 \left[1 - \left(\frac{P'}{P_1} \right)^{\frac{n-1}{n}} \right]$$

The work of the second stage is

$$W_2 = \frac{n}{n-1} P' V' \left[1 - \left(\frac{P_2}{P'} \right)^{\frac{n-1}{n}} \right]$$

Since P_1 and V_1 and P' and V' represent the pressures and volumes at

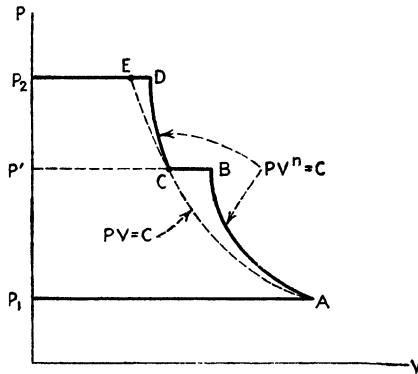



FIG. 130.—Theoretical cycle of two-stage compressor without clearance.

points A and C, respectively, and since these points are on an isothermal line, $P_1V_1 = P'V'$. Hence the total work is

$$W = W_1 + W_2 = \frac{n}{n-1} P_1 V_1 \left\{ 2 - \left[\left(\frac{P'}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_2}{P'} \right)^{\frac{n-1}{n}} \right] \right\}$$

The total work becomes a minimum when the following expression has a maximum value:

$$\left(\frac{P'}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_2}{P'} \right)^{\frac{n-1}{n}}$$

Differentiating the above expression with respect to P' and equating the derivative to zero, the value of P' is obtained which gives the maximum value of the expression and a minimum value for total work.

$$\frac{d}{dP'} \left[\left(\frac{P'}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_2}{P'} \right)^{\frac{n-1}{n}} \right] = 0$$

$$\frac{P'}{P_1} = \frac{P_2}{P'} = \left(\frac{P_2}{P_1} \right)^{1/2}$$

From this the total work calculated to be a minimum is:

$$W = \frac{2n}{n-1} P V_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{2n}} \right] \tag{100}$$

Example 7-6.—Calculate the horsepower of a two-stage compressor, compressing 230 cu. ft. of air per minute from 14.2 lb. per square inch absolute to 125 lb. per square inch absolute. The value of n is 1.28. What is the pressure in the intercooler?

Solution.

$$\text{hp} = \frac{2nP_1V_1}{(n-1)33{,}000} \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{2n}} \right]$$

$$= \frac{2 \times 1.28 \times 14.2 \times 144 \times 230}{0.28 \times 33{,}000} \left[1 - \left(\frac{125}{14.2} \right)^{\frac{0.28}{2.56}} \right]$$

$$= \frac{2 \times 1.28 \times 14.2 \times 144 \times 230}{0.28 \times 33{,}000} (1 - 1.268) = -35 \text{ hp.}$$

$$P' = (P_1P_2)^{1/2} = (14.2 \times 125)^{1/2} = 42 \text{ lb. per square inch absolute}$$

149. Compressed Air.—In power-plant operation, as in industry, there are needs for air under pressures higher than those provided by fans. The types of compressors in use are two: the reciprocating piston in cylinder, and the rotary compressor. Compressors may be steam driven, either direct connected, through a belt, or through gears, with either reciprocating steam engine or steam turbine as the prime mover. Other drives used include electric motor, gas or oil engine, and water wheel, either direct connected, or with belt or gear transmission. The most common type in power-plant use is the steam-engine direct-connected drive.

For low pressures, rotary or turbine-type blowers without valves are used. These are either single or multiple stage. These consist of impellers which rotate at high speeds in specially designed casings, and ordinarily deliver large volumes of air. There is no contact between the impeller blades and the casing. The pressure built up is due to the high velocity imparted to the air by the impeller, the velocity energy being partially transformed into pressure. Turboblowers may be of the multi-stage construction, capable of delivering pressures up to 100 lb. per square inch.

Allis-Chalmers makes a single-stage, geared turboblower with electric-motor drive. This runs at 6,450 r.p.m. with high efficiency and with constant discharge pressure throughout the range of discharge capacity. The impeller wheel is of the open radial-blade type and is made of cast high-tensile aluminum alloy. Single helical gears are

used, the pinion being of heat-treated alloy steel. The rated inlet volume ranges from 600 to 3,000 cu. ft. per minute for the different sizes.

The Roots-Connersville positive blower or rotary air compressor is designed for pressures up to 30 lb. per square inch. Two lobar impellers mounted on parallel shafts rotate in opposite directions. Contour and finish of these impellers are such that a clearance of a few thousandths of an inch is precisely maintained by a pair of accurately cut timing gears. No internal lubrication is required. Air is drawn in through the inlet, trapped between the impellers and the casing, and forced into the outlet.

The Allis-Chalmers rotary compressor is of the multi-cellular "sliding-vane" type. This consists of a cylindrical casing in which the cylindrical rotor, smaller in diameter than the bore of the casing, is arranged eccentrically. A number of radial slots are cut along the entire length of the rotor, into which fit the sliding vanes. As the rotor turns, the sliding vanes are thrown out by centrifugal force, forming a number of cells, each cell being contained between the surface of the rotor, two vanes and the casing wall. Due to the eccentricity of the rotor, the cells increase during each revolution from a minimum capacity to a maximum, and again to a minimum, thus producing the suction and pressure effect.

To prevent excessive friction loss and wear, the blades are not allowed to bear on the wall of the casing. Instead "floating rings" are employed, having an inner diameter lightly less than the casing bore. The blades bear on these rings only and cause the rings to rotate with the rotor and blades. The outer surfaces of the floating rings are separated from the casing by a slight clearance. Lubrication is effected by oil under pressure injected at each floating ring and at the ends of the rotor. These compressors operate with discharge pressure up to 10 lb. per square inch for the air-cooled types and up to 50 lb. per square inch for the water-cooled types. Speed ranges from 400 to 1,740 r.p.m. and capacities from 50 to 2,700 cu. ft. per minute.

The turbo-compressor, generally with steam-turbine or electric-motor drive, has a number of impellers, through which air flows in series. The velocity head in the air leaving each impeller is transformed into pressure head before entering the next impeller. The increase in pressure added by each impeller is 5 lb. per square inch depending upon the rotative speed. An example of a compressor of this type is the Ingersoll-Rand turbine-driven compressor. This machine has a capacity of 10,000 cu. ft. free air per minute to a dis-

charge pressure of 100 lb. per square inch at 4,700 r.p.m., driven by a 2,100-hp. turbine.

For high-pressure work, reciprocating piston compressors are used almost exclusively. The drive may be steam cylinder, internal-combustion cylinder or electric motor. The air cylinder may be single or double acting, single or multi-stage. For single-stage compression a single cylinder may be used, or two or more cylinders, arranged in parallel or at an angle, all delivering air at the same pressure. In multi-stage compression, the cylinders may be arranged in tandem, parallel, at an angle or a combination of these, and the air is successively compressed as it passes through from one cylinder to the

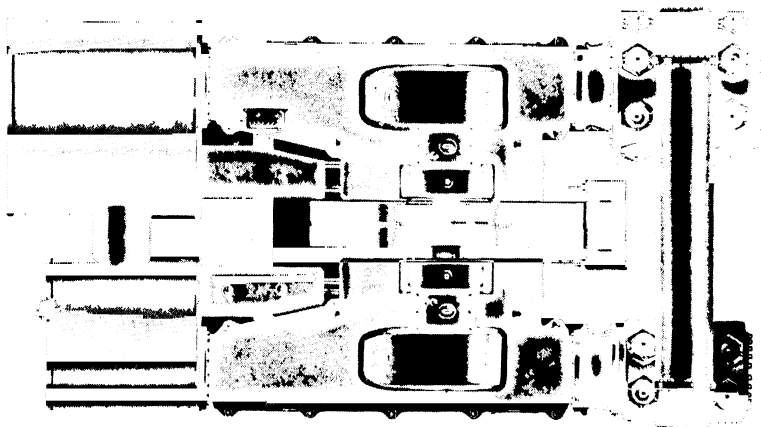



FIG. 131 –Plan view of Ingersoll-Rand XPV compressor.

next. Valves may be of the plate, feather, poppet, flap, Corliss, or slide-valve types.

It is common practice to make the discharge valves automatic in their operation. A spring holds the valve against the seat until the air pressure in the cylinder is sufficient to overcome the spring force and the pressure in the discharge pipe. The inlet valves are usually of the same type. This is not altogether true as many designs use positively operated inlet valves, either of the Corliss or slide-valve types. The automatic valves and seats are built in units, and can be installed or removed from the cylinder head completely assembled, thus facilitating repairing and replacing.

Plate valves consist of one or more thin steel discs covering smaller openings in the valve seat. These discs are held against the seat by small springs and are guided in their motion. Lift varies from 0.04 to 0.2 in.

"Feather valves" are made up of several thin strips of flexible material which fit into slots provided in one part of the seat. These strips cover openings in the other part of the seat and are held in position by their flexibility. With a lift of $\frac{1}{4}$ to $\frac{3}{8}$ in., the ends do not leave the seat, and the middle part bends upward against the guard above, the strip bowing upward. When seating there is no noise.

Poppet valves are usually cup or cone shaped and are provided with coil springs for controlling their operation. Several poppets are usually employed so as to keep down the size and weight of the individual moving part. A lift of $\frac{1}{4}$ to $\frac{3}{8}$ in. is common

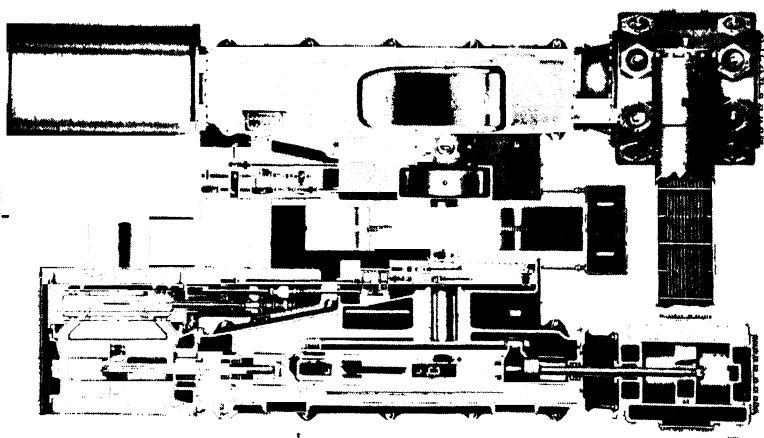



Fig. 132.—Top and sectional view of Ingersoll-Rand XPV compressor.

Flap valves consist of hinged plates which swing against the seat. They are held in closed position by a spring.

Figures 131 to 133 show the construction of the Ingersoll-Rand two-stage compressor type XPV with cross-compound steam cylinders. The flywheel is on the shaft between the two main bearings. The crosshead on the steam piston rod drives the crank shaft through the connecting rod. The crosshead on the air piston rod is connected to the steam crosshead by two rods, one on each side of the main connecting rod. The high-pressure steam cylinder and high-pressure air cylinder are on the same side.

The air valves are automatic, inlet and discharge. The valve consists of a grid seat, and a perforated steel disc, held in position by a guide plate, over the valve, which holds the springs and limits the lift to about $\frac{1}{4}$ in. A small pin keeps the valve from turning.

The intercooler, a long horizontal cylinder extends from the low-pressure to the high-pressure cylinder, being located above the air

cylinders. Air passes through the cooler between the tubes following a criss-cross path around baffles. Cooling water flows through the

FIG. 133.—Longitudinal sectional view of Ingersoll-Rand XPV compressor.

thin brass tubes in an opposite direction to the air flow. In practice, air is cooled to within 15° of the inlet water temperature.

TABLE 7-2.- COOLING WATER REQUIRED FOR TWO-STAGE COMPRESSION
(Gallons per 100 cu. ft. free air compressed to 80-100 lb.)

Temperature of water supply	60°	70°	80°	90°
Intercooler and jackets in series	2 9	3 4	4 0	4 5
Intercooler only	2 5	3 0	3 5	4 0
Cylinder jackets only	0.85	1 0	1 2	1 4

From Peele Compressed Air Plant

The steam cylinders, both high pressure and low pressure, use a design of Meyer cut-off valve. The main valve is a hollow slide valve into which the incoming steam enters. The exhaust steam leaving the cylinder passes around the ends. The riding cut-off valves, driven by a separate eccentric, have a reciprocating motion inside the main valve. The cut-off valves control the point of cut-off. These riding cut-off valves, one for each valve port, are right-hand and left-hand threaded, so the cut-off can be varied by turning the cut-off valve stem.

The governor is shown in Fig. 134. The oil pump delivers oil under pressure to the cylinder under the weight plunger. This oil pressure lifts the weight turning the cut-off valve sprocket which changes the point of cut-off in the steam cylinder. The oil pressure at the pump discharge is controlled by a pressure-regulating valve which when open permits oil to return to the reservoir thus

reducing the oil pressure. The position of this regulating or by-pass valve is controlled by the pressure of air from the high-pressure cylinder. Hence this governor runs the compressor not at constant speed but at constant pressure on the discharge air.

Care must be taken to prevent explosions in air-compressor cylinders. Especially in single-stage compressors, with compression up

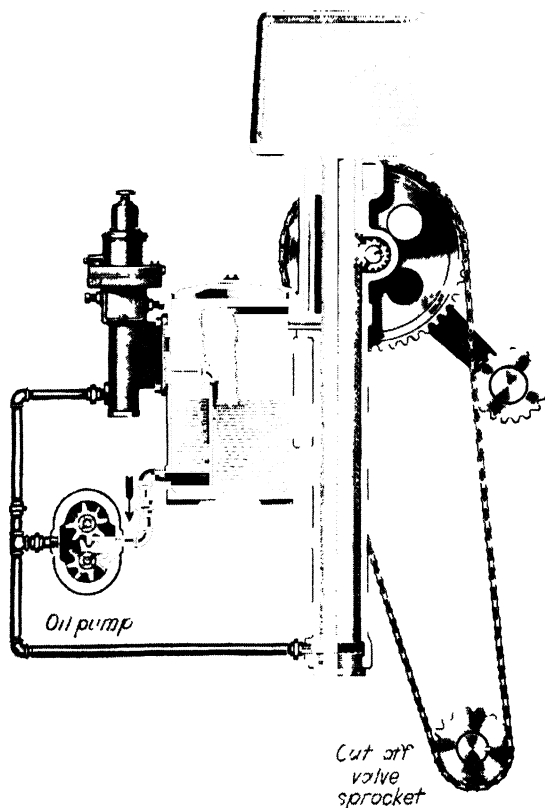



FIG. 134.—Diagram showing governor operation on Ingersoll-Rand XPV compressor.

to 100 lb., the air inside of the cylinder reaches temperatures close to 500°F. With lubricating oil of low flash point, the conditions are similar to those in an internal-combustion cylinder and there is danger of ignition. Only a high-grade, high-flash-point mineral oil should be used inside the air cylinder. The feed should be slow. The cylinders, pipes, and receiver tank should be cleaned out periodically, and scale formation in the jacket should be removed when it cuts down the effectiveness of the cooling.

In testing an air compressor the capacity or "free air" compressed must be measured. Free air is defined as dry air at atmospheric pressure and temperature as taken into the compressor cylinder, in cubic feet per unit time. The quantity of air delivered by a compressor may be measured by an orifice, low-pressure nozzle, Pitot tube, or displacement tanks. The most commonly used test devices are the low-pressure nozzle and the circular orifice with rounded approach.

TABLE 7-3.—COMPRESSOR SIZES AND CAPACITIES*

Cylinder diameter		Stroke	Speed	Capacity
Low-pressure cylinder	High-pressure cylinder		R.p.m.	Piston displacement, cu. ft. per min.
11	7	10	250	273
13½	8	10	250	411
15	9¼	12	235	574
17	10½	12	235	735
18	11	14	225	922
20	12½	14	225	1,140
23	14	16	200	1,530
23	14	20	180	1,723
26	16	20	180	2,200
30	19	24	155	3,026
32	20	30	125	3,465
35	22	30	125	4,150
36	23	30	125	4,400
41	26	30	125	5,700
44	27	30	125	6,575
46	28	30	125	7,180

* Courtesy of Ingersoll-Rand.

The compressor is commonly equipped with a receiver which serves as a reservoir, eliminating any pulsations, and to some extent cooling the air and reducing the moisture content. For testing with the low-pressure nozzle, a secondary receiver equipped with a nozzle of known dimensions is connected to the main receiver. The flow of air to the secondary receiver, and hence the pressure in this receiver, is controlled by a hand-operated valve. Provision is made to take the air temperature and pressure before entering the nozzle. The following equation by S. A. Moss gives the volume of air flowing through a

nozzle. This equation includes a correction factor changing the volume to the temperature at the compressor intake.

$$Q = \frac{2.552D^2CT_2}{P_2}\sqrt{\frac{BH}{T_1}} \qquad (101)$$

in which

Q = flow of air per minute at pressure P_2 and temperature t_2, cu. ft.
D = minimum diameter of nozzle throat, in.
C = coefficient of discharge, 0.98 to 0.99.
P_2 = pressure at compressor intake, lb. per square inch absolute.
T_2 = temperature at compressor intake, °F., absolute.
T_1 = temperature before nozzle, °F., absolute.
B = barometer reading (value of P_2), in. of Hg.
H = pressure drop through nozzle, in. of water.

The measurement of air flow through a circular orifice with rounded edges is based on Fliegner's formulas.

$$w = 31.8C\frac{P_1A}{(T_1)^{1/2}} \qquad \text{when } P_2 < 0.53P_1 \qquad (102)$$

$$w = 6.36C\frac{A}{(T_1)^{1/2}}[P_2(P_1 - P_2)]^{1/2} \qquad \text{when } P_2 > 0.53P_1 \qquad (103)$$

in which

w = flow of air per minute, lb.
C = coefficient of discharge, 0.99.
P_1 = pressure before orifice, lb. per square inch absolute.
P_2 = pressure in discharge region, lb. per square inch absolute.
T_1 = temperature before orifice, °F., absolute.
A = area of orifice, sq. in.

Problems

1. Determine the total draft loss, in inches of water, between the ash pit and chimney opening for a boiler unit equipped with natural-draft stokers when the following conditions apply: draft in furnace 0.75 in., loss through boiler 0.75 in., loss through flues and breeching 0.13 in., loss due to turns 0.15 in. No damper loss.

2. A boiler plant is to install four water-tube boilers equipped with forced-draft stokers and natural draft. The breeching between the chimney and the most remote boiler is 90 ft. long. There is one right-angle turn where the flue gases enter the breeching and one on entering the chimney. Required furnace draft is 0.17 in., and the boiler draft loss is 1.1 in. Determine the total available draft in inches of water required at the chimney.

3. Determine the total available draft required of the chimney of a boiler plant for the following conditions: draft over fire 0.07 in.; draft at boiler outlet 1.7; draft loss through damper 0.02.

Gases leave the boiler in a horizontal direction and enter the horizontal breeching in a vertical direction. Length of breeching is 60 ft.

4. Determine the maximum theoretical draft for a chimney 225 ft. high and 12 ft. in diameter. Prevailing barometric pressure is 27.5 in. of mercury; air temperature 75°F.; average temperature of flue gases 450°F.

5. Determine the maximum available draft of a chimney 185 ft. high and 8 ft. in diameter. Barometer 28 in. of mercury; air temperature 70°F.; gas temperature at chimney entrance 600°F.

6. Calculate the height of a chimney necessary to produce 1.85 in. of water theoretical draft. Temperature and barometer conditions as in Problem 4.

7. With the data given in Problem 1, determine the height of the chimney necessary. Temperature and barometer conditions as in Problem 5.

8. What are the required height and diameter of chimney to produce 2.25 in. of water theoretical draft, assuming the following conditions: pounds of coal per hour 15,000; 20-lb. waste gas per pound of coal; design gas velocity 25 ft. per second; air temperature 65°F.; gas temperature at chimney entrance 575°F.; barometer 25.5 in. of mercury; gas density as for air.

9. Solve Problem 8 if coal burned per hour is 18,000 lb. and gas velocity is taken as 30 ft. per second.

10. Determine the capacity, pounds of gas per hour, of a chimney 90 in. in diameter, for the following conditions: air temperature 70°F.; barometer 28.5 in. of mercury; average temperature of chimney gases 500°F.; $V = 20$ ft. per second.

11. Using Kent's formula determine the boiler horsepower capacity of a chimney 200 ft. high and 13 ft. in diameter.

12. Calculate the diameter, in feet, of a chimney 175 ft. high to serve 1,000 boiler hp. Use Kent's formula.

13. Static pressure in a fan duct is -2 in. of water; velocity pressure is 0.2 in. Determine the total pressure.

14. Calculate the average velocity (feet per minute) of air, at 80°F., flowing in a circular duct if the center velocity pressure is 0.6 in. of water. Center coefficient is 0.91 and barometer 28 in. of mercury.

15. Air at 90°F. flows in a duct at an average velocity of 30 ft. per second. Calculate the average velocity pressure in inches of water if the barometer reading is 28.2 in. of mercury.

16. A delivery of 20,000 cu. ft. of air per minute, at 2.5 in. static pressure, is required of a fan having a mechanical efficiency of 65 per cent. The duct through which the air flows is 4 ft. sq. If the density of the air discharged is 0.069, determine the air horsepower of the fan, and the brake horsepower required to drive it.

17. A fan takes air at 70°F. from a long duct 2 ft. sq. and discharges it into a duct slightly larger. The maximum average velocity pressure is 0.2 in. of water; inlet static pressure is -0.19 in. of water; discharge static pressure is 5 in. of water; barometer 27 in. of mercury. Determine (*a*) the quantity of air, cubic feet per minute, flowing, (*b*) air horsepower, (*c*) brake horsepower required to drive the fan if the mechanical efficiency is 55 per cent.

18. Boiler test data (note all quantities on 1-hr. basis):

Coal, as fired: Ultimate analysis, per cent, M 1.51, C 73.44, H 4.76, O 6.28, N 1.45, S 0.71, ash 11.85; coal 10,183 lb.; ash and refuse 1,500 lb.; carbon in ash and refuse 20 per cent.

Flue-gas analysis, per cent volume: CO_2 12.9, O_2 6.1, CO 0.06.

Temperatures, °F.: air to furnace and room 79, air surrounding chimney 40, gas entering chimney 300, average chimney gas 240.

Miscellaneous data: barometer 28.6 in. Hg, gas velocity in chimney 30 ft. per second, area of fan duct 15 sq. ft.

a. Calculate theoretical draft for a chimney height of 150 ft.

b. Calculate height of chimney to give an available draft of 0.7 in. of water.

c. Calculate chimney diameter (for dry gas only) if 3 boilers use one chimney.

d. Same as part *c*, above, except that 6 boilers use one chimney.

e. How many boilers, operating the same as above data, could a chimney of 18 ft. diameter serve (dry gases only)?

f. Calculate air horsepower of a forced-draft fan serving one boiler, static pressure 2.5 in. water.

g. Same as part *f* for an induced-draft fan serving 2 boilers.

19. Coal data, as fired: Alabama, Bibb County (page 53); coal, pounds per hour, 25,500; ash and refuse, pounds per hour 1,655; carbon in ash and refuse, per cent, 20; flue-gas analysis, per cent: CO_2 12.5, O_2 6.3, CO 0.5.

One chimney serves 5 boilers. Outside air 30°F., average temperature gas 400°F., barometer 30. Calculate: (*a*) chimney diameter for a gas velocity of 25 ft. per second; (*b*) height of the chimney for a theoretical draft of 1.5 in. water.

20. A fan supplied the boiler of Problem 19 with combustion air, total static pressure 5 in. water, area of air duct 15 sq. ft. Calculate the air horsepower.

21. *a.* Air is drawn into an air compressor at a temperature of 60°F. and at a pressure of 14.2 lb. per square inch absolute. The flash point of the oil used to lubricate the compressor piston is 500°F. If the compression is adiabatic what pressure could be attained in the compressor with the maximum allowable temperature 50°F. below the flash point of the oil?

b. During adiabatic expansion the temperature of air falls from 600 to 20°F. Find the ratios of $p_2 : p_1$ and $V_2 : V_1$.

22. What will be the difference in the amounts of work necessary to compress 8 cu. ft. of free air (air at 60°F. and 14.7 lb. per square inch absolute) to a pressure of 90 lb. per square inch absolute when the compression is adiabatic and when it is isothermal?

23. Find the power required to compress 1,200 cu. ft. of air per minute in a two-stage compressor according to $PV^{1.35}$ = constant. $P_1 = 14.2$ lb. abs., $P_2 =$ 155 lb. abs., $t_1 = t' = 150$°F.

24. A compressor takes in 250 cu. ft. of air per minute at 14.7 lb. per square inch absolute and compresses it to 176.4 lb. per square inch absolute according to the law $PV^{1.25} = c$. Find the horsepower required if the compressor is (*a*) single stage, (*b*) two stage.

25. An air compressor is to compress 300 cu. ft. of air per minute from 14.3 lb. per square inch absolute to 143 lb. per square inch absolute. Assuming conditions ideal, (*a*) what will be the pressure in the intercooler? (*b*) with $n = 1.3$, what will be the saving in work due to two staging?

26. A single-stage double-acting air compressor, 15 by 18 in., 180 r.p.m., piston rod 1¾ in., clearance volume 5.6 per cent of piston displacement, compresses air according to $PV^{1.35}$ = constant. $p_1 = 14.4$ lb. abs., $p_2 = 95$ lb. abs. Determine volumetric efficiency and the horsepower of compression.

27. An air compressor with a cylinder 12 by 18 in. is double acting and is driven at a speed of 100 r.p.m. Required the capacity of the compressor in cubic feet of free air per minute. $n = 1.33$, clearance is 1.9 per cent; $p_1 = 14.5$ lb. per square inch absolute; $p_2 = 60$ lb. per square inch absolute.

Determine the work necessary to compress 800 cu. ft. of free air per minute from a pressure of 14.7 lb. per square inch absolute to a pressure of 110 lb. per square inch gage: (*a*) isothermally, (*b*) polytropically, $n = 1.33$.

28. A multi-stage air compressor receives air at 60°F. when the barometer reads 29.5 in. mercury and raises it to 56 lb. per square inch gage in the first stage. The air is then cooled to 60°F. In the second stage the pressure is raised from 56 lb. gage to 220 lb. gage. All compression adiabatic. Find the work per pound of free air drawn into the compressor.

29. An air compressor with a cylinder 20 by 30 in. is double acting and is driven at a speed of 80 r.p.m. Clearance is 3 per cent of piston displacement; $p_1 = 14.4$ lb. per square inch absolute and $p_2 = 75$ lb. per square inch absolute. Find the horse power required to compress the air. (Take $n = 4/3$ and atmospheric pressure $= 14.7$ lb. per square inch.)

CHAPTER VIII

STEAM SUPERHEATERS AND SEPARATORS

150. Development in the Use of Superheated Steam.—The practice of *superheating* steam for use in prime movers was introduced during the early part of the nineteenth century. At that time reciprocating steam engines were the only prime movers of any practical importance. Steam pressures of 40 lb. per square inch were considered high and unsafe, and the cylinder lubricants available were incapable of withstanding temperatures much in excess of 300°F. Consequently, progress in the application of superheated steam was greatly limited, and it was not until mineral lubricants and improved materials came into general use that any real notice was given to it.

The development of the steam turbine, over the last 30 years, has, perhaps, been the greatest factor affecting the advancement in superheating practice. The progress of the two has, for the most part, been inseparable. With the increased use of turbines, superheaters gradually became a necessary part of the plant equipment, and at present they are installed in almost all of the power plants of any importance throughout the world. Superheated steam temperatures of 600 to 1000°F., at pressures of from 700 to 1,400 lb. per square inch, are now being used with safety, and manufacturers are prepared to go even higher.

151. Advantages of Using Superheated Steam.—The advantages of the use of superheated steam are many in number, and they depend on factors relating, principally, to the nature and character of the plant equipment. Chief among the advantages are considerable increases in both the economy and efficiency of the prime movers, boilers and steam piping. These are incident mainly by virtue of the increased specific volume and available energy and the gaseous nature of steam when superheated.

When steam expands adiabatically in a turbine a part of its heat is necessarily abstracted, and, if it is initially dry and saturated, moisture particles are formed. Under these conditions the blades deteriorate rapidly, and the loss due to friction between the steam and the turbine parts becomes excessive, especially in the lower stages. It is therefore necessary to supply turbines, chiefly the larger ones, with superheated steam in order to reduce these effects to a minimum.

In extremely large turbines it is often impractical to supply steam sufficiently superheated to ensure dry steam in the last stages. In such cases the steam is bled from one of the lower stages and resuperheated or reheated. It is then returned to the turbine and further expanded to the exhaust pressure.

When used in reciprocating engines, superheated steam will reduce or completely eliminate cylinder condensation, one of the chief losses, thereby lowering the heat and water consumption. A considerable amount of heat may be abstracted from superheated steam before condensation occurs, which is not true of saturated steam. Also, the heat units per unit of volume are less, and from this a heat saving is readily apparent, since a reciprocating steam engine uses a definite volume of steam for each stroke. In general, steam turbines show less increase in steam economy, due to superheating, than do reciprocating engines. Direct-acting pumps probably show the largest gain.

Superheaters attached to boilers nearly always increase both the boiler heat-producing capacity and the overall efficiency. By virtue of its smaller volume per B.t.u. theoretically available, superheated steam may be transported through pipes of reduced size, compared to saturated steam. Radiation loss will be reduced by the lower thermal conductivity. Elaborate drain equipment and the tendency for a water hammer, which are incident with saturated steam, are also eliminated.

Along with the many advantages of the use of superheat steam, it must be noted that they may be offset to some extent by the increased expenditure for equipment. Considering everything, however, there nearly always results a net financial gain wherever it is used.

152. Superheater Types.—The principle involved in superheating steam for power use consists of passing the steam, after leaving the boiler proper, through the heated tubes or passages of a superheater. The superheater is installed in a way so as to be exposed to either hot gases or fire or partially to both. The method of heat absorption forms the basis for a classification of superheaters, thus dividing them into two types:

1. Convection.
2. Radiant.

Either or both types are installed, with the boiler, in the boiler setting, in which case they are termed *integral* or *attached*. If they are installed and fired apart from the boiler setting they are called *separately fired* or *direct-fired* superheaters.

Each superheater is made up of a number of looped tube units, the ends of which connect with suitable inlet and outlet headers. The

inlet header, in most cases, receives steam direct from the main steam drum of the boiler while the outlet header connects with the service line.

153. Integral Superheaters.—Integral *convection* superheaters are usually installed in or between the first and second passes, when

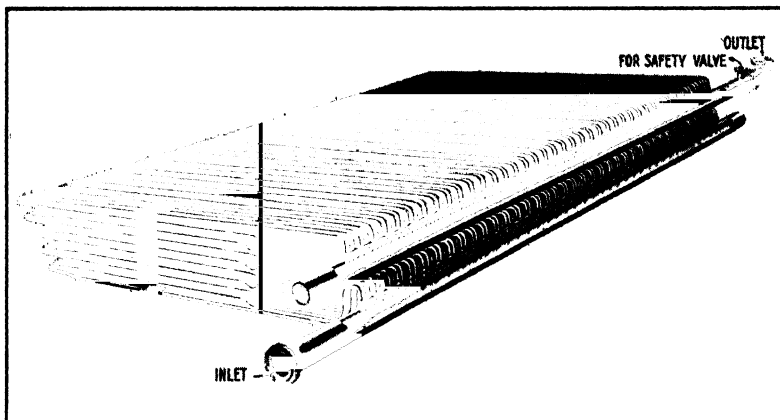



Fig. 135.—Elesco convection superheater.

used with water-tube boilers, and they are conveniently arranged so as to be protected from the furnace heat by one or more rows of water tubes. Such an arrangement is illustrated in Figs. 2, 31, 32, 34, 40, 41 and 69.

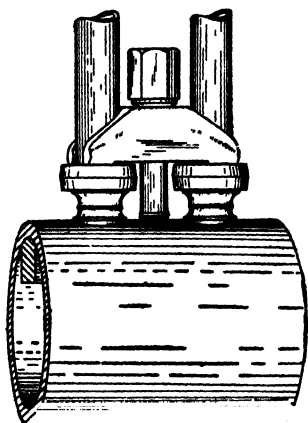



Fig. 136.—Showing method of attaching Elesco superheater elements to headers.

Figures 42 and 44 (pages 104 and 106) illustrate installations of the *Elesco superheater,* manufactured by The Superheater Company and very widely used. The superheater, alone, is shown in Fig. 135. Each tube element is formed separately and attached to the headers by clamps, as shown in Fig. 136. The type of construction renders this superheater conveniently adaptable to almost any shape and style of boiler.

Elesco *radiant superheaters* are made from the same tube elements. In this case, however, they are often shaped so that they form a plane surface when assembled in the furnace. They may be directly exposed to the furnace heat, or partially protected by water-wall tubes.

The Foster *convection superheater* is illustrated in Fig. 137. The chief distinctive feature of this superheater is the cast-iron ferrules

which are machined and shrunk onto the steel tubes carrying the steam. These serve as a protection against corrosion and excessive heating and give an added element of strength. The general construc-

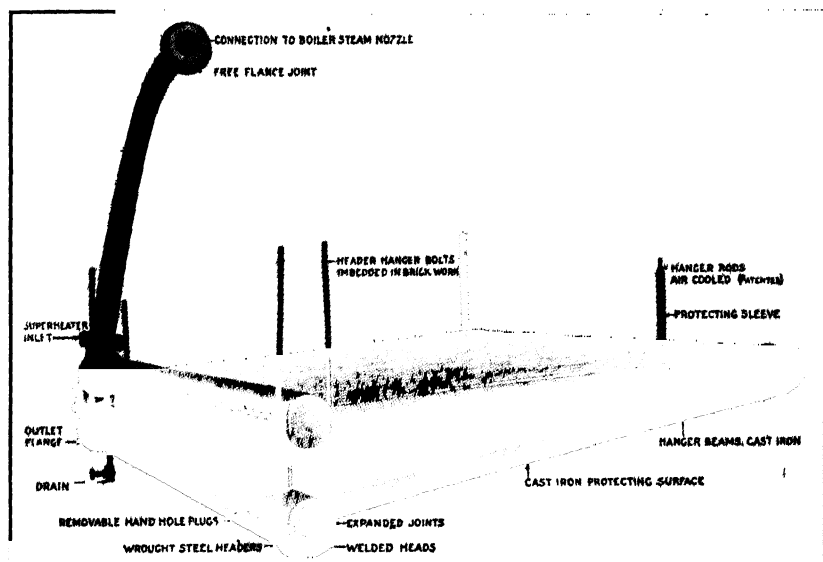



FIG. 137.—Foster convection superheater.

tion details are brought out in Fig. 138. The typical Foster elements are U shaped, as shown in Fig. 137. They are, however, made straight, where the U bend is not practical, giving the assembled superheater the appearance of a gridiron.

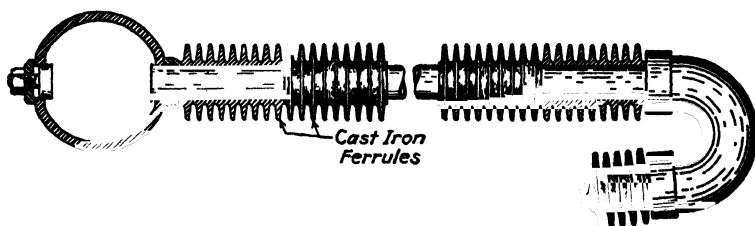



FIG. 138.—Showing construction of Foster superheater element.

Provision is made within the superheater to aid in the distribution of the steam to the various elements. This is accomplished by the use of either orifice plugs or pipe cores which are closed at their ends and held central within the tube elements by lugs. The pipe cores form an annular space for the passage of steam, which, in addition to restricting flow, greatly aids the superheating process.

The construction of a Foster *radiant superheater* element is illustrated in Fig. 139. The headers are the same as those used in the convection type, but the heating elements are rectangular, forged-steel boxes, connected to the headers, as shown in the figure, by tube

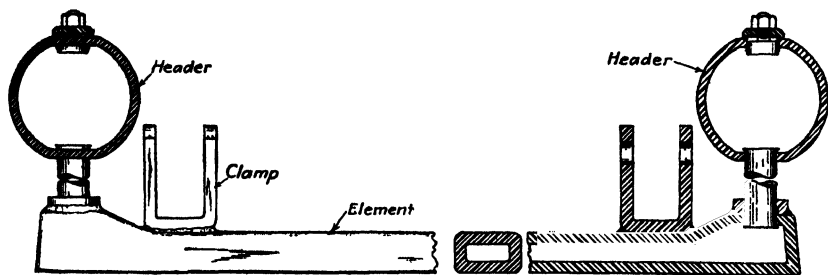



FIG. 139.—Foster radiant superheater element.

nipples. The elements are assembled side by side, either horizontal or vertical, forming a flat and exposed wall surface when the superheater is installed in the furnace.

The Babcock and Wilcox *convection superheater* is, in some respects,

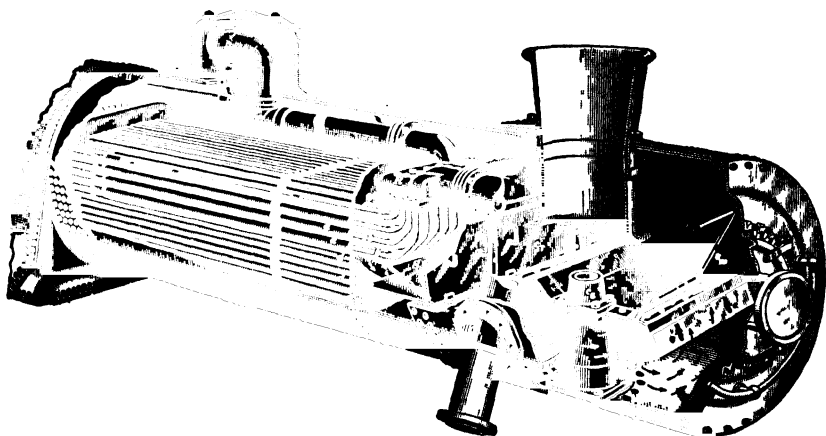



FIG. 140—Showing a locomotive superheater installed (*Courtesy of International Textbook Company*)

similar to the Elesco. Long tubes of small diameter and short bends are compactly grouped. Figures 31 and 34 (pages 92 and 95) showing the superheater installed, give a general idea of its construction.

Superheaters are widely used in modern *locomotive* practice. By reason of the construction of a locomotive boiler, the superheater elements are built to extend into especially large flue tubes (*a*, Fig. 21, page 84) in the upper tube area. Installed thus, they may be called *extended* or *flue-tube superheaters*. Figure 140 shows a typical instal-

lation of a locomotive superheater. The headers *R* are located in the smoke box, at the front end of the boiler, and the elements *b* extend into the large flue tubes *a*, as shown, each making two complete loops or four passes before reaching the outlet header. Saturated steam is delivered from the throttle valve in the steam dome, through the outlet pipe to the superheater. After passing through the superheater elements, the superheated steam flows direct to the valve chests of the engine cylinders. As a protection against excessive heating of the elements when there is no steam passing through the superheater, an automatically operated damper (*S*, Fig. 140) is provided to stop the flow of hot gases through the superheater flues. A small cylinder and piston (not shown), located at the side of the boiler and connected with the steam supply pipe, with the aid of levers, is the means used for actuating the damper.

154. Separately Fired Superheaters.—Separately fired superheaters are used for the following reasons:

1. When the boiler design does not permit the installation of an integral superheater.
2. When the steam is to be superheated at a considerable distance from the boiler.
3. When the steam is to be highly superheated (750 to 1250°F.).
4. When the steam from one boiler or a battery of boilers is to be used for a variety of purposes and at various temperatures.

They consist of a series of tube elements suitably placed within a brick or metal setting and having the headers usually on the outside. With a suitable furnace, almost any kind of fuel may be used as the source of heat, and, when possible, considerable economy is effected by the use of waste heat from some industrial process. Separately fired superheaters have the advantage of greater flexibility and more accurate control, but they are more expensive and require additional attention.

155. Superheater Performance.—Superheater design and performance are dependent on a large number of factors which are, in some cases, exceedingly variable and latent. Consequently, the design and installation of superheaters are based largely on experience. In the case of integral superheaters, the problem is mainly one of determining the proper amount of superheating surface and the placement of the superheater to carry the load and give the desired steam temperature, for various conditions of boiler operation. In the boiler setting, convection and radiant superheaters differ considerably in their performance. This difference is apparent on referring to the curves in Fig. 141, which were plotted from data obtained during a

large central-station boiler test. With an increase in percentage of boiler rating, the temperature rise in the convection superheater gradually increases to a certain point, after which there is a slight reduction. The opposite is true of the radiant superheater. By using both types and placing them in series, the variable effect of each is lessened and the resultant superheat is more or less constant, within

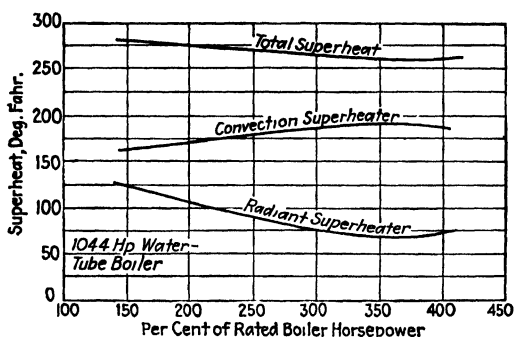



Fig. 141.—Integral superheater performance curves.

the range of ordinary operation. With suitable relations between the two superheaters, the total-superheat curve becomes nearly horizontal.

Superheater surface may be calculated by the following equation:

$$UAD_m = W_s(h_2 - h_1) \tag{104}$$

in which

U = heat absorbed, B.t.u. per hour, per square foot of surface, per degree Fahrenheit.
A = superheater surface area, sq. ft.
D_m = mean temperature difference, °F.
W_s = steam per hour, lb.
h_2 = enthalpy per pound at exit, B.t.u.
h_1 = enthalpy per pound at entrance, B.t.u.

$$D_m = \frac{t_2 - t_1}{\log_e \dfrac{t_3 - t_1}{t_3 - t_2}}$$

t_2 = temperature of steam at exit, °F.
t_1 = temperature of steam at entrance, °F.
t_3 = average temperature of gas surrounding superheater, °F.

156. Reheat and Desuperheat.—It is common practice to *reheat* or *resuperheat* steam between the cylinders of large multiple-expansion steam engines and between the lower stages of large steam turbines. In this way, additional available energy is added to the steam, and

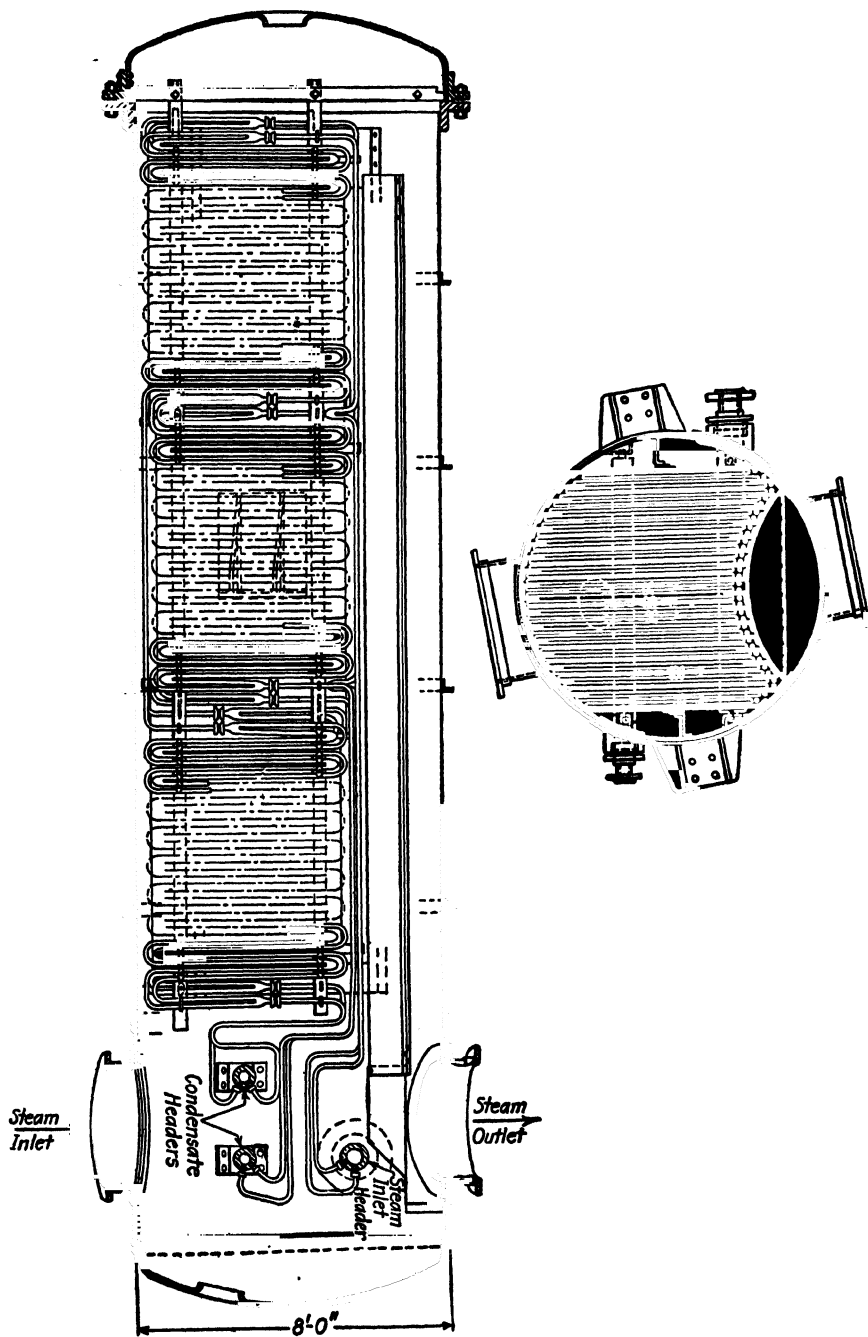



FIG 142.—Elesco steam reheater.

more work is obtained from it in the course of expansion from a high initial pressure to a final exhaust pressure.

Reheating may be accomplished by the use of separately fired, integral or live-steam superheaters. The two latter are most common in the power field. Integral convection reheaters, located in the gas passes of the boiler setting beyond the initial or primary superheaters, are often used in large-turbine practice. Live-steam reheaters had their early existence with multiple-expansion engines, and recently they have been applied to turbines with remarkable success.

In a *live-steam reheater*, the steam to be reheated is passed through a closed steel cylinder where it absorbs heat by contact with tubes containing live steam at high pressure and temperature. To minimize the piping and radiation loss it is placed as near as is possible to the prime mover. A reheater of this type is shown in Fig. 142. This reheater is built in various sizes, to handle from 20,000 to 200,000 lb of steam per hour.

It is often necessary to *desuperheat* steam when the pressure and temperature at which it is generated are too high for a particular purpose. This is accomplished by spraying water into the path of the steam or by the use of closed coolers. The closed cooler is, in principle, similar to the preheater shown in Fig. 142, except that a cooling agent, such as water, is used instead of the steam. In this way a desuperheater is sometimes used as a feedwater heater.

STEAM SEPARATORS AND DRYERS

157. Steam Separators.—Steam separators are used for the purpose of removing entrained water and moisture particles from saturated-steam lines. Water carried into turbines or reciprocating engine cylinders is rapidly destructive to the working parts and greatly lowers the efficiency and economy. General sluggishness of the whole system is also one of the effects of wet steam.

The principal causes of moisture in steam are priming within the boiler and condensation in the steam lines. Generally speaking, separators are unnecessary with superheated steam. In such cases, however, they are usually installed as a means of safety in starting up. Also, there is always a danger of superheaters becoming flooded and slugs of water being carried into the steam lines.

The principles by means of which moisture may be separated from saturated steam are based on (1) centrifugal force, (2) reverse current, (3) the use of baffle plates, and (4) the use of a contact mesh. In the first two cases, the principle depends on the momentum of the moisture particles, while in the last two it is their property of adhering to sur-

faces. Figure 143 shows a horizontal type of steam separator which utilizes a baffle and reverse current. Some of the moisture particles are thrown against and adhere to the baffle, and the collected moisture flows to the lower part of the chamber. A large portion of the moisture falls to the bottom of the chamber, as illustrated, as the steam reverses its direction of flow. The vertical separator shown in Fig. 144 utilizes centrifugal force. A whirling motion is given to the steam as it passes through the spiral vanes in the inlet. The moisture particles are sup-

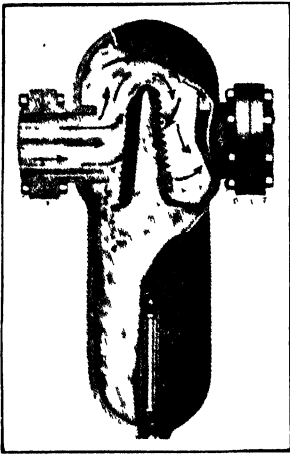



Fig. 143.—Wright-Austin horizontal separator.

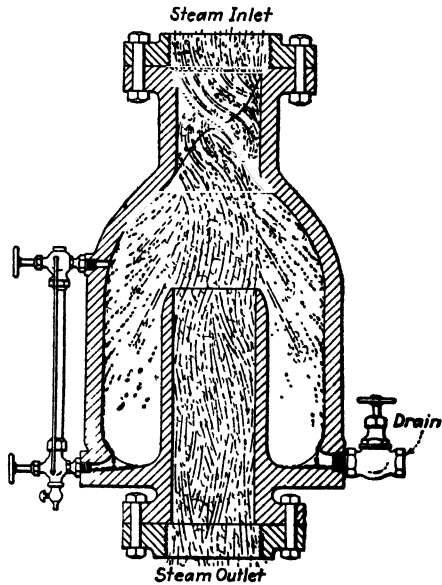



Fig. 144.—Swartout vertical steam separator.

posedly thrown against the inner wall surface and collect at the bottom of the separator.

Separators should be drained automatically, by traps. Otherwise, frequent attention is required if they are to serve their purpose.

158. Steam Dryers and Washers.—Dry steam is one of the requisites of a good boiler. A common device to produce dry steam is the "dry pipe" (see Fig. 25, page 88). The dry pipe is about the same diameter as the boiler outlet nozzle and is placed close to the top of the steam drum. The top side of the pipe is perforated and the steam rising from the water surface makes a 180-deg. turn to enter the holes. At low and moderate ratings this prevents moisture from entering the nozzle.

The Badenhausen boiler (Fig. 35, page 96) is equipped with heating pipes through which the steam must pass after it leaves the water

surface. In this passage it undergoes a drying or even a superheating action from the furnace gases.

In other boilers baffles are provided extending almost the width of the drum. The steam must pass at low velocity around these baffles, and moisture separates from the vapor in this passage.

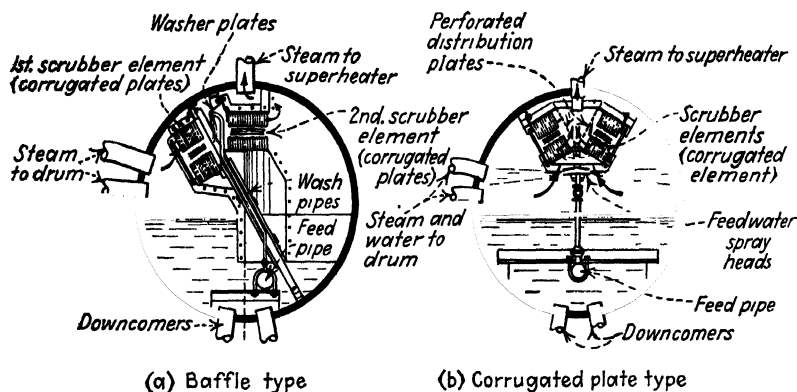



FIG. 145.—Steam washers and dryers.

Steam washers and dryers (Fig. 145) are provided to eliminate the carry over of solids in the steam. Steam is first washed by entering feedwater which has a lower solid concentration than the water in the boiler, and hence dilutes the concentration of solids in the water entrained with the steam. The washed steam is then dried by corrugated plates or heavy screening before passing to the steam outlet.

CHAPTER IX

FEEDWATER HEATING AND TREATMENT

159. Feedwater Heating.—Feedwater is prepared for the boiler by heating or chemical treatment, or both. Feedwater heaters have two functions, (1) to heat water, and (2) to purify water. The heat may be supplied by steam (exhaust or high pressure) or by the waste gases leaving the boiler. In the first case the equipment is known as a *feedwater heater*, and in the second case, as an *economizer*. This use of exhaust steam or waste gases effects a gain in plant efficiency by returning to the boiler heat which otherwise would be wasted. Another factor to be considered is the improvement in the condition of the boiler. The use of hot feedwater eliminates severe strains that would be set up if cold water came in contact with hot boiler metal.

One of the most important results of heating water lies in the increased evaporation obtained from a boiler. In modern central stations where large turbines are used, the water is heated in successive steps so that it enters the boiler at very close to the saturation temperature of the boiler steam, and the heat absorbed in the boiler by each pound of water is very little more than the latent heat of vaporization. Plants operating on this cycle, the *regenerative cycle*, heat the water by means of a series of open and closed heaters supplied with steam bled from different stages of the turbine unit. In practice, the regenerative cycle produces higher plant efficiencies than any other cycle where steam is the sole medium for heat transfer.

Heaters have a beneficial effect on boiler water by removing part of the scale-forming impurities and by driving off certain dissolved gases which may be harmful to boiler metal. In many small plants, the water treatment used is that effected by the open feedwater heater.

160. Open Feedwater Heaters.—The open feedwater heater is one in which the steam and water come into direct contact, being different from the closed heater in which there is metal between the steam and water. A typical open heater is illustrated in Fig. 146. It is divided into an upper heating chamber and a lower storage chamber or hot well, which also contains a filter bed. A float-operated regulating valve maintains the hot-well water level constant.

Cold water enters a distributing tray at the top and cascades over a series of staggered trays, the edges of which are serrated to increase

the water surface exposed to steam. Steam first passes through a multiple-baffle oil separator, then into an annular space around the sides of the heater chamber, flowing downward and entering the tray nest at the bottom. Rising through the finely divided spray of water, the steam gives up its heat of evaporation, condenses, and falls back to the hot well with the heated water. If the cold feedwater contains temporary hardness, a considerable part of the soluble salts is precipi-

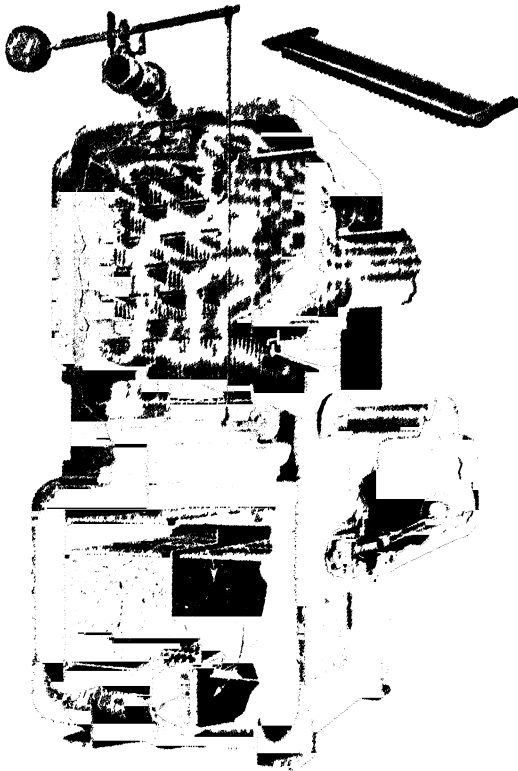



Fig. 146.—Worthington open feedwater heater.

tated, when heated, and is deposited as scale on the trays. Through doors, these trays are readily accessible and easily cleaned. A vent to remove non-condensable gases is provided in the top of the heater. At temperatures existing in an open heater, air, oxygen and carbon dioxide gas become less soluble. Consequently, the open heater will remove a considerable part of the dissolved gases from water. The overflow trap at the side collects the drip oil from the oil separator and discharges it to waste. This also serves as an overflow in case of

high water. The filter bed contains coke or other filtering material, and is baffled so that the heated water flows upward through one side and downward through the other side, as shown, to the suction pipe leading to the boiler feed pump. Doors are provided for cleaning or renewing the filtering material.

Figure 147 shows a sectional view of an open feedwater heater having a metering chamber. The heated water flows into a weir chamber and over a V-notch weir into the hot well. By means of the float which controls the recording pen (Fig. 148), the flow over the weir is indicated and integrated. A pen records the flow upon a clock-driven chart having uniformly spaced divisions. A large pointer

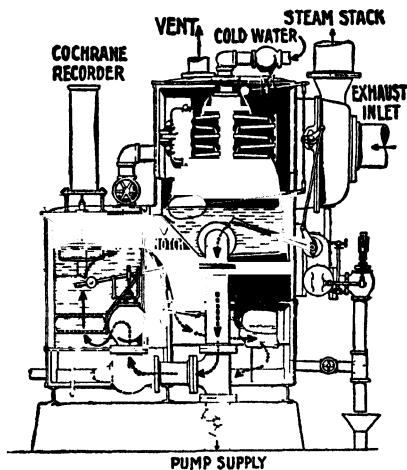



Fig. 147.—Cochrane open feedwater heater, with recording meter.

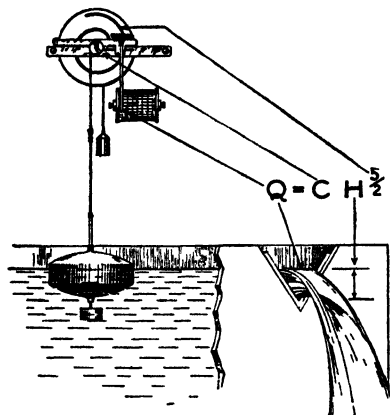



Fig. 148.—Illustrating the principle of the V-notch meter

shows the momentary rate of flow and the accumulated flow is added by a counting train. With the exception of the weir chamber and metering mechanism this heater (Fig. 147) is similar to the one previously explained. This type of heater is used where low pressure steam is used for heating coils or process work, in factories. All of the exhaust steam goes through the oil separator, then a valve set by hand diverts a part up the stack to the process lines. The valve is of the rotary type and can be set to divert any part or all of the steam up the stack.

Figure 149 illustrates the horizontal cylindrical type of open feedwater heater, which, in its principle of operation, is like the one shown in Fig. 147. The upper space is filled with trays arranged to divide the water into finely divided cascades in contact with the steam, and the lower part is the storage space. The water level is kept constant

by a float which operates a valve on the cold-water supply line. A variable amount of water is received from the pumped return line. As this is distilled water it is admitted, freely, to the heater.

In a jet heater (Fig. 150) the water and steam are in direct contact. This, therefore, may be classed as an open-type heater. The heating medium, usually low-pressure steam, flows into the cylindrical chamber, and the cool water is sprayed radially into this chamber through a row of nozzles placed around the periphery. As a result, the water is very finely divided and is effectively heated.

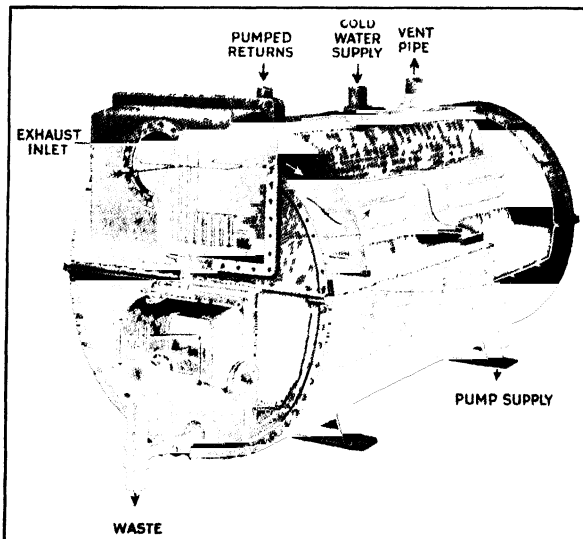



FIG. 149. –Cochrane horizontal open feedwater heater.

161. Closed Feedwater Heaters. –In heaters of this type the water and steam are not in contact, but the heat is transmitted from the steam to the water through the walls of metal tubes. Generally, the water is inside the tubes. The steam is outside the tubes, and as it condenses the condensate is drained, by gravity, to a trap. In efficient plants, the trap drains from the closed heaters are returned to the feedwater circuit, to a surge tank, or to a low-pressure heater. This reduces the loss of pure water and, to that extent, decreases the amount of make-up water that must be supplied to the circuit. The water being heated may be at any required pressure, and the pressure of the steam used for heating may be at any available pressure, depending on the water temperature desired. Closed heaters are often used in series for heating the feedwater in successive steps.

Impure water or water containing oil forms a coating similar to scale on the inside of the tubes of a closed heater. Such a coating reduces, considerably, the heat transmission and the efficiency of the heater. Cleaning the tubes of this coating is difficult, and for this reason in practice this type of heater is used for water which contains no scale-forming impurities.

Fig. 150.—Cochrane jet heater.

Figure 151 shows the construction of a horizontal, floating head, closed type of heater, built to withstand exceedingly high pressures and temperatures. Water enters and leaves the heater at the same end, and it makes four passes through the straight tubes. Steam enters the heater shell at the top; and the tubes immediately below the inlet are protected, by a baffle, from the erosive action of the steam on entering. To allow for tube expansion and contraction at one end, the floating head, which is free to move along the shell, is employed. At the bottom of the heater is the float-operated valve for automatically removing the condensed steam.

Fig. 151.—High pressure (1,200-lb.) closed heater. (*Foster Wheeler Corporation.*)

Figure 152 shows the tubes and header assembly for a hairpin or U-tube closed heater. This tube design is used in vertical heaters

and is well adapted for high temperatures and pressures. Each tube is free to expand, individually, and there is no stress placed on the header due to unequal temperature changes in the different parts of the heater. The bundle of tubes can be easily removed from the heater for inspection or repair.

162. Calculations for Water Heaters.—In water heaters there is a flow of energy from the high temperature or heating medium, such as

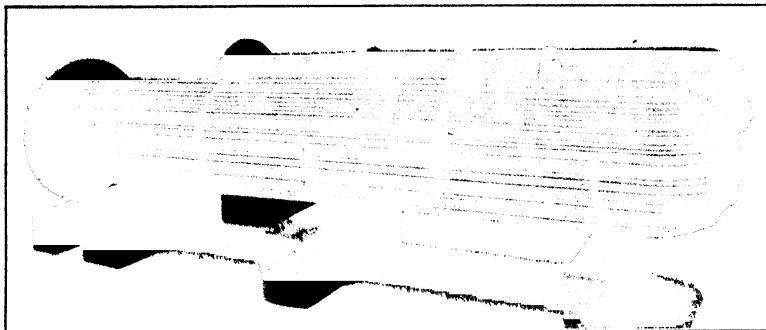



Fig. 152.—Tube bundles of hairpin or U-tube closed heaters.

steam or gases (gases with economizers), to the water being heated. If radiation losses are assumed so low as to be negligible, the energy flow may be expressed, simply, as heat absorbed by the water or as the heat rejected by the steam or gas. In the case of the closed heater or economizer this equals the heat transmitted through the metal surface of the tubes. These relations are expressed by the following equations:

$$Q = W_w(h_{f2} - h_{f1}) = W_s(h_3 - h_{f2}) \quad \text{(open heater)} \tag{105}$$

$$Q = W_w(h_{f2} - h_{f1}) = W_s(h_3 - h_{f3}) = AUD_m \quad \text{(closed heater)} \tag{106}$$

$$Q = W_w(h_{f2} - h_{f1}) = W_g \times .24(t_5 - t_4) = AUD_m \quad \text{(economizer)} \tag{107}$$

in which

Q = total energy flow, B.t.u. per hour.

W_w = weight of water entering heater (or economizer), lb. per hour.

W_s = weight of steam entering heater, lb. per hour.

W_g = weight of gas entering economizer, lb. per hour.

t_1 and t_2 = water temperatures, entering and leaving heater (or economizer), respectively, °F.

t_4 and t_5 = gas temperatures entering and leaving economizer, respectively, °F.

p_3 = steam pressure, lb. per square inch absolute.
t_3 = saturation temperature of steam to heater.
U = coefficient of heat transmission, B.t.u. per hour per square foot of surface per degree Fahrenheit temperature difference (see Table 9-1, for heaters).

For economizers U may be used as from 4 to 10, with an average of 7 B.t.u. There are many factors that will change the value of U. Among these are gas velocities, condition of metal surfaces, thickness of metal, conductivity, and water velocities.

D_m = mean temperature difference between steam (or gas) and water, °F., where

$$D_m = \frac{t_2 - t_1}{\log_e \dfrac{(t_3 - t_1)}{(t_3 - t_2)}} \qquad \text{(for closed heater)}$$

$$D_m = \frac{(t_4 - t_2) - (t_5 - t_1)}{\log_e \dfrac{(t_4 - t_2)}{(t_5 - t_1)}} \qquad \text{(for economizer with counterflow)}$$

TABLE 9-1.—VALUES OF U, COEFFICIENT OF HEAT TRANSMISSION[1]

Type of heater	Velocity of flow, ft. per min.	B.t.u.
1. Multiple flow heaters	50	
Plain copper tubes	....	250
Corrugated copper tubes	....	300
2. Single flow heaters	12.5	
Plain brass tubes		175
3. Coil pipe heaters	150	
Plain copper tubes	....	300
Plain iron tubes	....	120

[1] Values from "Mechanical Equipment of Buildings," by Harding and Willard.

Example 9-1.—A single-flow closed heater with plain brass tubes heats 550,000 lb. of water per hour from 206 to 268°F. The steam temperature is 290°F. What area is theoretically required?

Solution.

$$D_m = \frac{(t_2 - t_1)}{\log_e \left(\dfrac{t_3 - t_1}{t_3 - t_2}\right)} = \frac{(268 - 206)}{\log_e \left(\dfrac{290 - 206}{290 - 268}\right)} = 46.3\text{°F.}$$

$$A = \frac{W(t_2 - t_1)}{UD_m}$$

$$= \frac{550{,}000(268 - 206)}{175 \times 46.3} = 4{,}200 \text{ sq. ft.}$$

163. Purpose of Economizers.—The main evaporating effect in the boiler occurs in the first pass, where the hottest gases first come in contact with the tubes. After the first pass, the heat transmission decreases as the temperature difference between water and gas becomes smaller, until the last pass is reached, which, in the average boiler, is comparatively inactive. The theory of the economizer is to reduce the heating surface of the boiler, so that the gases leave the boiler at a

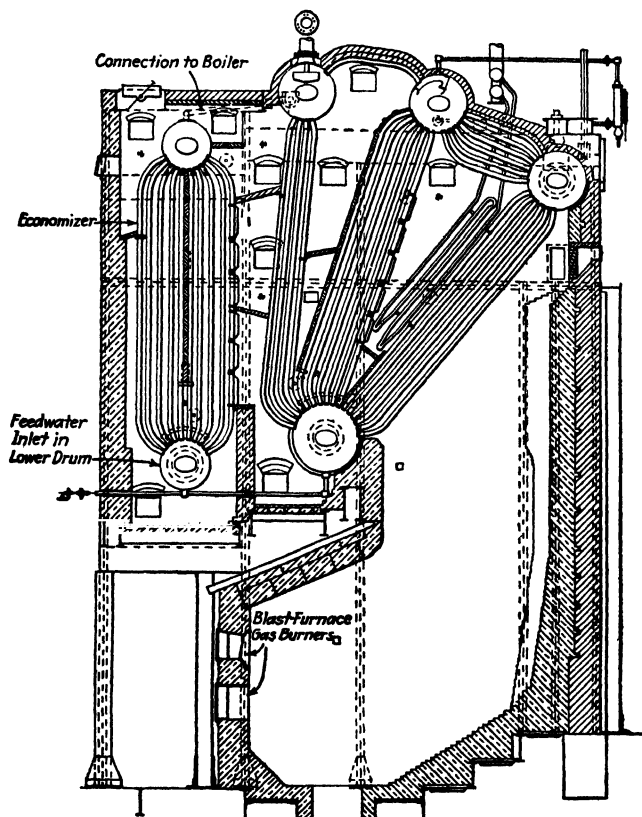



Fig. 153.—Connelly boiler with integral economizer.

comparatively high temperature and enter the economizer where additional heat is effectively transmitted to the water. This is due to two factors, (1) that the temperature of the water in the economizer is less than that in the boiler, and (2) that there is a countercurrent flow of gas and water, both of which maintain a high temperature difference. In many installations the gases are cooled so effectively by the economizer that they enter the stack at a lower temperature than that of the water in the boiler.

164. Economizer Types and Construction.—Boilers similar to the Stirling or Badenhausen often have an economizer element as part of the boiler. These are called *integral type or steaming economizers.* An economizer of this type is illustrated in Fig. 153.

The *independent type of economizer* is one that is ordinarily placed apart from the boiler setting proper. In the usual construction the tubes are horizontal, with countercurrent flow relation of flue gas and water. As the tube nest forming the economizer offers considerable

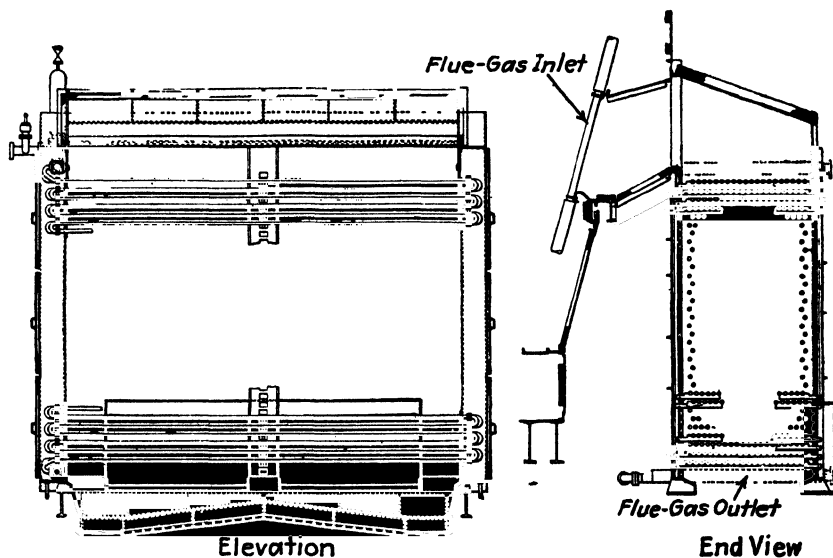



Fig 154 —Babcock and Wilcox economizer.

resistance to the gas flow, an induced-draft fan is placed between the economizer and breeching.

The *Babcock and Wilcox economizer* is shown in Fig. 154. There are two headers, one at the top and one at the bottom. Between, is a series of rows of horizontal 2-in. tubes, supported, one row above the other, in a staggered position, with each tube in each row connected at either end to the tubes of the next row, above or below, by a flanged return bend. The water leaves the inlet header and is divided into as many parallel streams as there are tubes entering the header. This parallel flow is continued through a series of horizontal tubes until the last rows of tubes discharge the water into the outlet header.

The return bend construction is shown in Fig. 155. The bends are of seamless steel tubing, with the ends threaded, faced and fitted with flanges. The ends of the economizer tubes are upset, threaded and fitted with similar flanges. In assembling, a thin gasket is used

between the tube and bend, and the flanges on each are held by two bolts. The cast-iron support for the tubes is also illustrated in Fig. 155. This support has a second function; namely, that of protecting the two headers and the many return bends from the hot gases. Each tube passes through a counterbored hole in the support, and a thin soft-steel split ring is placed around the tube and pushed against the shoulder of the counterbore. The ring serves to center the tube and to protect the braided asbestos rope packing which is driven

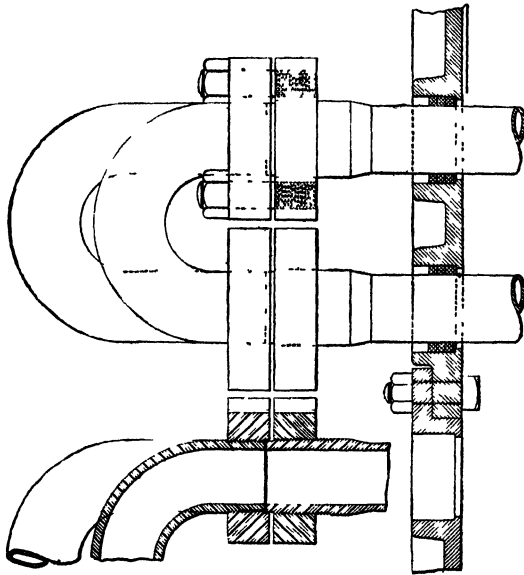



Fig. 155.—Showing return bend construction, Babcock and Wilcox economizer

into the hole around it. This forms a gas- and water-tight joint which eliminates any possibility of corrosion at any of the joints between tubes and return bends. Along the vertical edges of the tube end supports, handholes are provided for inspection or removal of tubes. These holes are equipped with tightly fitting covers held in place by clamps. As shown in Fig. 154, the economizer tubes are supported at the center by flat rolled-steel bars, with recesses milled out to receive the tubes.

Hoppers are provided at the bottom to collect soot that is blown from the tubes. The sides are surrounded by a casing of built-up cast-iron panels or sheet-metal plates. Usually these are insulated. Doors are provided to give ready access to the tube joints.

The Foster economizer, installed as shown in Fig. 156, has several construction details that are different from other economizers. The

tubes (Fig. 157) are 2-in. steel tubes on the outside of which is tightly fitted a series of cast-iron gilled rings, similar to the cast-iron ferrules

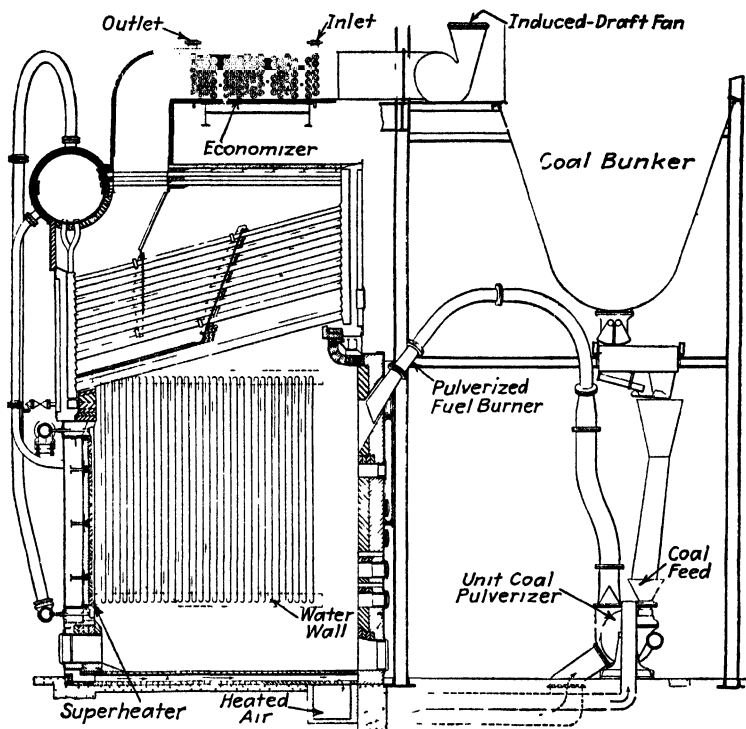



Fig. 156.—Section view showing installation of Foster economizer.

used on Foster superheaters. The ends of these ring castings are provided with male and female joints so that the steel tube is com-

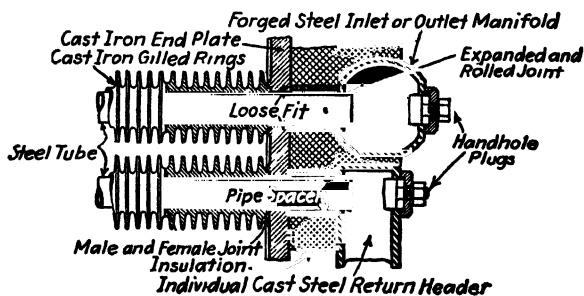



Fig. 157.—Showing tube and header construction, Foster economizer.

pletely covered. Hence, the tubes are protected as only the cast-iron covering is exposed to the hot gases. Cast iron is a better material than steel for resisting corrosion due to the presence of sulphurous

acid in the flue gas. Another advantage of this design lies in the large amount of heat-absorbing surface offered by the corrugated rings.

Each tube is connected to the tube just above by means of a forged-steel return header (Fig. 157). Cleaning of the inside of tubes is effected through handholes which are closed by handhole plugs held in position by a yoke and nut. The pressure within the header holds the plug tight against the seat. The tubes are supported at the ends, and in the case of long tubes, at the center, by cast-iron supports.

Soot blowers are commonly installed for cleaning the outside surface of economizer tubes. When out of service the tube surface is washed, thoroughly, with water. The soot is collected in the bottom, from where it is removed.

165. Results of Poor Boiler Water.—Using a poor quality of water in the boiler will result in one or more of the following conditions:

1. Priming and foaming.
2. Corrosion.
3. Caustic embrittlement.
4. Scale formation.

Foaming is the production of a mass of frothy bubbles in the steam space of the boiler. This is caused by the gradual concentration of soluble sodium salts and is aggravated by the presence of suspended solids, oil, grease or organic matter. Generally speaking, boilers begin to foam when the water concentration reaches 200 to 300 gr. of sodium sulphate per gallon. *Priming* describes the condition in the boiler when slugs of water are delivered with the steam, and it usually occurs in a boiler that is foaming. Priming and foaming may cause damage to superheater tubes or to steam engine or turbine parts.

Corrosion consists of the attacking of boiler metal by acids in the water or by electrolytic action. Corrosion is taken up in detail in a subsequent paragraph.

Caustic embrittlement is the term applied to the condition of boiler metal in which small hair-line cracks appear, usually in highly stressed regions of the steel. The metal in the affected parts loses its toughness and becomes quite brittle. This condition is possible with water containing a considerable amount of caustic soda (NaOH). The accepted theory is that the caustic soda attacks the iron, liberating hydrogen which is adsorbed or occluded by the iron, bringing about the change in its physical properties.

Scale formation on boiler heating surfaces is due chiefly to the decreasing solubility of certain slightly soluble salts with increases in temperature. Initial deposits of scale crystals may occur by sedimen-

tation or by trapping in surface depressions. The cause of the initial deposit and the subsequent growth of scale in the boiler may be a supersaturated solution with respect to the incrustants in water. Scale once formed undergoes a baking effect producing a solid layer. Chemical analyses of scales indicate the most common constituents to be calcium sulphate, magnesium hydroxide, magnesium silicate, and calcium silicate, with calcium carbonate and calcium hydroxide occasionally present. Sulphates have a cementing effect on the scale layer, while silica scales have a characteristic of toughness.

The result of scale, of course, is to decrease the heat conductivity. Values of U, the B.t.u. transmitted per square foot of heating surface per hour per degree Fahrenheit temperature difference, for scale vary from a minimum value of 0.05 for very porous scale to 1.3 for dense scale of calcium sulphate, with a maximum value of 2.

166. Chemical Treatment.—Chemical treatment of water has for its purpose purification to prevent one or more of the troubles listed in the preceding article. The treatment may be either external, that is, before the water enters the boiler, or internal, as in the case where boiler compounds are used The method chosen depends upon local conditions.

All natural water, except distilled or fresh rain water, contains foreign substances, such as dissolved salts, gases, or suspended matter. Under the conditions of high pressure and temperature certain of the salts become insoluble and form a coating of scale on the metal surface of the boiler. As this coating becomes thicker the heat transmission is proportionately reduced.

Acids in water have the opposite effect, attacking the metal either over a wide area (general corrosion) or locally in small spots (pitting). Corrosion or pitting appears to be more active if dissolved oxygen is present in the boiler water. An excessive amount of alkaline salts in water may result in caustic embrittlement.

In large plants, careful and continuous chemical analyses of the feedwater are made, and expensive equipment for exact chemical treatment is frequently installed. This expense nearly always proves economical, as it eliminates unnecessary shutdowns as well as prevents serious damage to boiler equipment.

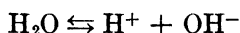
167. Corrosion.—Corrosion may be a general eating away of boiler metal to a slight depth, or it may be localized, showing up in small circular pits of considerable depth, the latter being known as pitting. There are many factors that enter into corrosion. Consequently, in many cases it is difficult to determine the cause and apply the correct remedy. Any water containing free acid is corrosive, and during

corrosion an acid solution will evolve hydrogen gas on the surface of the metal.

Magnesium chloride or magnesium sulphate will cause corrosion. Under boiler conditions, these salts react with water to form hydrochloric or sulphuric acid. Dissolved gases, principally oxygen and carbon dioxide, also play an important part in corrosion. In natural waters, corrosion is almost directly proportional to the oxygen concentration. Except in rare cases, complete degasification of average boiler water will eliminate corrosion.

Electrolytic action may be the cause of corrosion; *i.e.*, some foreign material acting as one pole and a spot of pure iron on the boiler surface as the other. With the slight galvanic action, the positive pole will be eaten away gradually, and, at the negative pole, hydrogen will be evolved. The hydrogen oxidizes to water by the dissolved oxygen.

Pure, distilled water is composed of H_2O molecules and also of hydrogen (H^+) and hydroxyl (OH^-) ions, the exponents representing positive and negative electrical charges. The processes of splitting up of molecules into ions, and the recombining of ions into molecules, continue until there is an equilibrium as shown by the following equation:

$$H_2O \leftrightarrows H^+ + OH^-$$

This indicates that the splitting and recombining effects are exactly equal so that the total number of ions and the total number of molecules remain constant. A measure of the acidity, or H^+ ions present, give an indication as to the corrosiveness of boiler water and is expressed by the pH coefficient.

The pH coefficient designates the fraction of a gram of hydrogen ions present in each liter of water. The figure gives the number of ciphers in the denominator. Thus pH4 would indicate that there is 1/10,000 g. of hydrogen ions in one liter of water, while pH7 would designate 1/10,000,000 g. of hydrogen ions per liter. A water with pH7 value is considered neutral. Lower pH values denote higher acidity, and higher pH values show an alkaline condition.

In reporting the degree of acidity in terms of hydrogen-ion concentration, the results are in terms of the acid ions present in the solution, and not in terms of total acid or alkali, as in the case with titration methods. Hydrogen-ion concentration thus becomes a measurement of quality or activity of the solution just as a B.t.u. determination is a measure of the quality of a given weight of coal. It is possible to have a water which is alkaline to titration methods, but, as shown by the

hydrogen-ion determination, has acid properties which cause it to be corrosive.

Surface waters are nearly always saturated with oxygen. Natural waters also contain carbon dioxide present in solution as H_2CO_3,

TABLE 9-2.—THE RELATIONSHIP BETWEEN HYDROGEN AND HYDROGEN-ION CONCENTRATION AND pH NUMBER

Nature of solution	Concentration of hydrogen and hydroxide ions, grams per liter		pH number
Acid, hydrogen ions predominate	1.0	Hydrogen ions (H^+)	0
	0.1		1
	0.01		2
	0.001		3
	0.0001		4
	0.00001		5
	0.000001		6
Neutral, hydrogen and hydroxide ions present in equal quantity	0.0000001	Hydrogen (H^+) and hydroxide (OH^-) ions	7
Alkaline, hydroxide ions predominate	0.000001	Hydroxide ions (OH^-)	8
	0.00001		9
	0.0001		10
	0.001		11
	0.01		12
	0.1		13
	1.0		14

which ionizes slightly, thus increasing the hydrogen ionization and lowering the pH value. Counteracting this, most waters contain bicarbonates of calcium and magnesium which tend to increase the pH value. The following relation exists between the bicarbonate hardness, free carbon dioxide and the pH value for natural waters:

$$\text{pH} = \frac{\text{alkalinity as } CaCO_3 \text{ in p.p.m.} \times 2}{CO_2 \text{ in p.p.m.}} \times 10^6$$

A water low in carbonate hardness is likely to be more corrosive than one of relatively high carbonate hardness. The use of alum as a coagulant, reducing the carbonate hardness and increasing the free carbon dioxide, will increase the corrosive tendency of water.

The presence of hydrogen ions in high concentration increases the corrosive effect, as this indicates high acidity. Such water should have a proper amount of alkali added to bring it to a neutral or alka-

line condition as represented by a high pH value. Reagents which are used to correct an acid condition include sodium, calcium or barium hydroxide.

To remove oxygen from water two types of apparatus are utilized, (1) *de-activator* and (2) *de-aerator*. The de-activator, rarely used, consists of a storage tank filled with a labyrinth of iron sheets. The water passes through this tank and acts corrosively on the iron therein. This uses up the dissolved oxygen and discharges water freed from this gas.

More important is the de-aerator, the principle of which is illustrated in Fig. 158. Boiler water enters a closed heater and flows by gravity through a float-regulated valve into the separator tank where

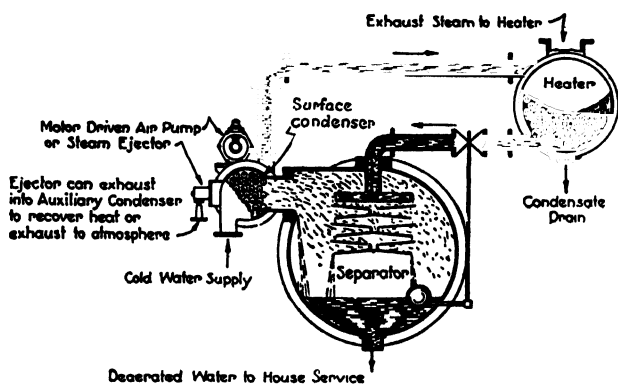



FIG. 158.—Diagrammatic section drawing of an Elliott de-aerator.

a reduced pressure exists. The water cascades over staggered heating tubes and partially flashes into vapor, releasing the dissolved and entrained non-condensable gases. An air ejector creates a vacuum and pulls the vapor and gases upward through the surface condenser, as shown. Here the vapor condenses and drains back to the bottom of the separator chamber. The air passes through the ejector, and is discharged to the atmosphere, direct, or after passing through an auxiliary condenser (not shown). The auxiliary condenser may serve to condense the steam from the ejector and facilitate returning its heat and water to the feedwater circuit. The boiler water passes through the tubes of the auxiliary and main condensers, in series, before going to the heater, thus saving the heat that would otherwise be wasted.

Corrosion in boiler tubes may be reduced by allowing a thin coating of scale to form on the inside surface. It is occasionally the

practice, after cleaning the tubes, to use untreated water for a short time to obtain this protective coating.

When off the line for any length of time, a boiler should be dried by a light furnace fire of wood. Lime should then be placed in the boiler drums to maintain dryness of the interior surfaces.

Zinc slabs may be placed in boiler drums to prevent electrolytic action on the boiler metal, during operation.

168. Caustic Embrittlement.—Caustic embrittlement is the term applied to the condition of boiler metal in which small hairline, intercrystalline cracks appear. These cracks occur below the water line and are irregular in direction. They are not joined and are found in highly stressed regions, particularly around riveted joints. Embrittled metal generally shows the presence of black iron oxide and is very brittle under shock. Ductility may be at least partially restored by heating or by removing the boiler from service for a time. Investigators, among whom are Prof. F. G. Straub and W. C. Schroeder, have concluded that the presence of caustic soda (NaOH) and sodium silicate (Na_2SiO_2) in the boiler water is the principal cause of embrittlement. Cracking takes place if the sodium hydroxide content is 100 p.p.m. and the silica content 0.6 p.p.m. If the silica content is less, but other impurities, including CaO, MgO, and Al_2O_3 being present in such quantities as to make up for the lack of silica, embrittlement is also likely to take place. Chemically pure sodium hydroxide, alone, does not bring about embrittlement, but, with the presence of the other agents in suitable quantities, embrittling action occurs.

The prevention of embrittlement is accomplished best by maintaining the A.S.M.E. recommended sulphate-total-alkalinity ratio, as follows:

Boiler Pressure, Lb. per Sq. In.	Ratio of Sodium Sulphate to Total Alkalinity as Equivalent Sodium Carbonate
Up to 150	Not less than 1 to 1
150 to 250	Not less than 2 to 1
Over 250	Not less than 3 to 1

These ratios call for increasing concentrations of sodium sulphate, which, on the other hand, may result in calcium scale formation, corrosion, or foaming.

Feedwater used in embrittled boilers has been found to be high in sodium bicarbonate and low in sodium sulphate. Upon heating, the sodium bicarbonate breaks down to give the carbonate, which in

turn gives sodium hydroxide and carbon dioxide as shown in the reaction equation

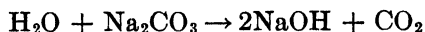
$$H_2O + Na_2CO_3 \rightarrow 2NaOH + CO_2$$

In *Straub's* investigations, boilers have been found in which over 90 per cent of the sodium carbonate has changed to sodium hydroxide. The prevention of embrittlement has been effected by eliminating the formation of sodium hydroxide in the boiler. The addition of magnesium sulphate forms sodium sulphate, thus decreasing the sodium carbonate proportion. The addition of sufficient sulphuric acid to neutralize a major portion of the sodium carbonate content has also proved an effective preventive. Other methods require such reagents as aluminum sulphate, iron sulphate, etc., all using the same principle, namely of maintaining a certain definite amount of sodium sulphate in boiler waters which contain sodium carbonate.

TABLE 9-3.—ANALYSES OF TYPICAL WATERS, GRAINS PER U. S. GALLONS

Sample	Steam pressure, lb. gage	Source of water	NaOH	Na_2CO_3	Na_2SO_4	NaCl	Na_2CO_3	$\frac{Na_2SO_4}{\text{Total alkalinity as } Na_2CO_3}$	Recommended A.S.M.E. ratio
A	200	Lake	85.4	69.2	44.7	27.3	184.0	0.24	2.0
B	250	River	24.3	10.5	36.2	24.2	44.5	0.81	3 0
C	30	Well	25.6	25.5	13.3	33.9	61.1	0.21	1.0
D	...	Well	22.0	9.0	90.0	....	38.0	2.4	2.0
E	...	Lake	36.3	14.6	119.7	....	55.4	3.1	3.0
F	...	Well	14.1	17.0	72.8	....	36.6	2.1	2.0

The method of calculating the total alkalinity is explained in Art. 170 on Water Analyses. The total alkalinity for sample *A* of the foregoing tabulation is determined as follows (see page 271):

NaOH $85.4 \times {}^{17}\!/_{40} = 36.2$ gr.

OH $36.2 \times 2.941 = 107$ gr. hydroxyl alkalinity

Na_2CO_3 $69.2 \times {}^{60}\!/_{106} = 39.2$ gr.

CO_3 $39.2 \times 1.667 = 65.5$ gr. carbonate alkalinity

$(107 + 65.5) \times 1.06 = 172 \times 1.06 = 184$ gr. total alkalinity as Na_2CO_3

Samples *A*, *B* and *C* (Table 9-3) were of waters which caused an embrittled condition in the boiler, while samples *D*, *E* and *F* were from boilers which had no trouble with embrittlement.

The new welded-drum boiler construction should decrease the possibility of embrittlement since there are no joints to offer contact to concentrated caustic solutions.

169. Evaporators.—An important factor in feedwater treatment is the use of distilled water for make-up. The cost of an installation for this purpose is low, and the quality of the distillate is high, even with poor grades of raw water. In certain cases where a very poor grade of water is available, chemical treatment of the raw water before

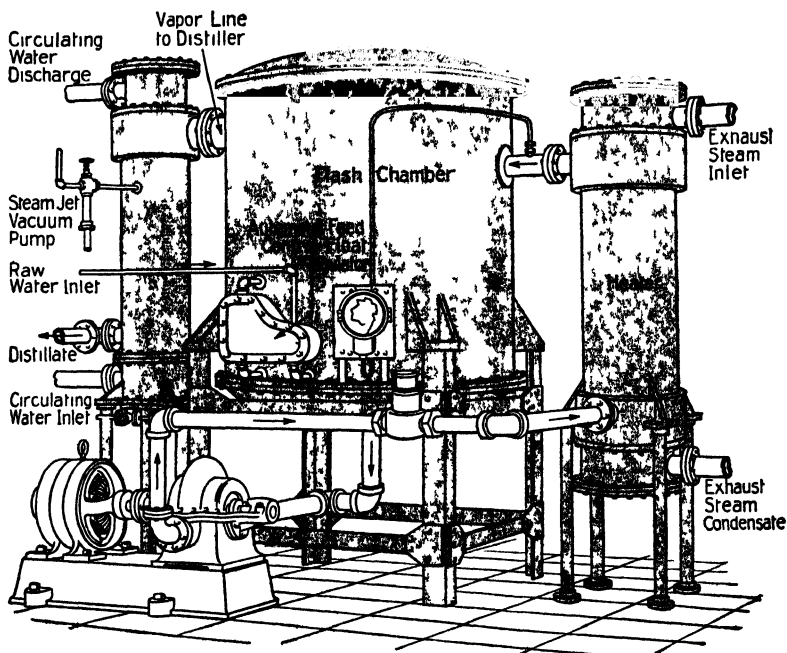



FIG. 159.—Schematic drawing of the Koerting flash evaporator.

it enters the evaporator is found to be economical. This additional expense is more than offset by the elimination of such effects as reduced capacity of the boilers and shutdowns for cleaning. However, in the ordinary evaporator installation the use of chemicals is unnecessary.

Evaporators may receive their heat from high pressure or exhaust steam, or from steam bled from a turbine. They may be classified according to the manner in which the water is vaporized as: (1) *flash type*, (2) *film type*, (3) *submerged type*.

In the *flash-type evaporator*, illustrated in Fig. 159, the raw water is heated in an exhaust-steam closed heater to about 200°F. The flash chamber is under a vacuum of about 16 in. mercury and is arranged so as to cause incoming water to divide into many thin

films. Consequently, a spontaneous and violent boiling occurs, and a part of the water flashes into steam. The vapor thus formed is then drawn into the condenser where it condenses, forming the distilled make-up water. The water level in the flash chamber is maintained constant by the automatic feed-control float regulator. The evaporator condenser may be of either the surface or jet type, using the condensate from one of the prime-mover units for the cooling water. A vacuum pump ordinarily is used to maintain the vacuum in the flash chamber.

The *film-type evaporator* consists of a shell enclosing a nest of tubes which carry high-temperature steam. The raw water is admitted to the hot well on the bottom of the shell and is pumped to the top from

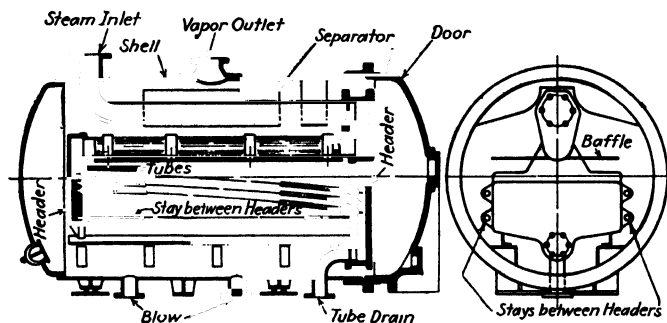



FIG. 160.—Griscom-Russell "Bentube" evaporator.

where it falls, in films, over the hot tube surface. The vapor formed is removed and condensed, giving the required distilled water.

The *bentube evaporator* (Fig. 160) may be operated as either a film-type or a submerged-type evaporator. The tubes are slightly bowed to assist in cracking the scale when the tube temperatures are changed rapidly in the scale-removing operation.

In the *submerged type of evaporator*, tubes, filled with steam, are entirely submerged by the raw water. The water, when evaporated, is removed and condensed as before.

Multiple-effect evaporators consist of evaporator units in series; that is, the vapor from one effect forms the heating medium for the next one. The usual multiple-effect installation consists of two or three evaporators, with the successive vapor pressures decreasing. In practice as many as 12 evaporating effects have been used.

Multiple-effect systems are classified as *low heat level* or *high heat level* evaporators, the main difference being the temperature of the boiler feed water produced. Two-effect evaporators of these two classes are shown, diagrammatically, in Fig. 161. It will be noted

that the raw water in both cases is heated in an open heater and then pumped to the evaporators, through a float-regulated valve. As the vapor in each unit condenses, the condensate is drained by a trap and discharged to a line leading to the boiler feedwater heater.

In the operation of an evaporator two troubles usually associated with boilers are experienced, namely, concentration and scale. Scale is removed automatically by quickly changing the tube temperatures, causing a cracking up of the scale. Excessive concentration is prevented by blowing off a certain amount of water from time to time.

(a) Low heat level. (b) High heat level.

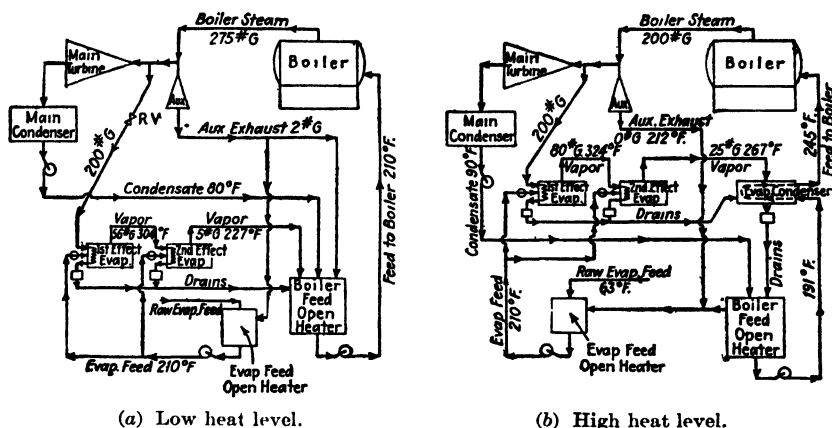



FIG 161 —Diagrams of two-effect evaporator systems.

This is usually taken care of by emptying the evaporator once each 24 hr. during the scale-removal process.

170. Water Analyses.—Water analyses are generally reported in terms of grains per U. S. gallon (equivalent to parts per 58,341) or in parts per million (p.p.m.). To change from one basis to another the following relations are used:

7,000 gr. = 1 lb (avoirdupois)
1 cu. ft. of water weighs 62.31 lb. at 70°F.
1 gal. (U. S.) = 0.134 cu. ft.
1 gal (U. S.) = 0.134 × 62.31 × 7,000 = 58,300 gr. at 70°F.
1 gr. per gallon = 1 gr. per 58,300 gr.
1 gr. per 58,300 gr. = 17.12 gr. per 1,000,000 gr.

Hence an analysis given in parts per million (p.p.m.) can be converted into the equivalent analysis in terms of grains per U. S. gallon by dividing each item, p.p.m., by 17.12.

The actual determination of the dissolved substances in water is usually made of the various elements or radicals, from which the combined analysis is calculated to show the probable combined salts.

Such an analysis will show the incrustants, non-incrustants, total suspended solids, and commonly the organic matter, all expressed either as grains per U. S. gallon or as parts per million.

The *Cochrane Corporation* uses the following rules in forming a combined analysis when given in the ionic form:

Case 1. *When the* $(HCO_3)_2$, CO_3, *and* SO_4 *radicals equal or exceed the* Ca *or* Mg *requirements, combine the solids in the following order:*

1. *Change the bicarbonate* $(HCO_3)_2$ *to its equivalent* (CO_3) *radical.*
2. *Magnesium* (Mg) *with carbonate* (CO_3).
3. *Remaining carbonate* (CO_3) *with calcium* (Ca).
4. *Remaining carbonate* (CO_3) *with sodium* (Na).
5. *If carbonate* (CO_3) *is insufficient for calcium, combine the remaining calcium with sulphate* (SO_4).
6. *Remaining sulphate* (SO_4) *with sodium* (Na).
7. *Chloride* (Cl) *with sodium* (Na).
8. *Nitrate* (NO_3) *with sodium* (Na).

Case 2. *When the* $(HCO_3)_2$, (CO_3), *and* (SO_4) *radicals are less than the* Ca *or* Mg *radicals, combine the solids in the following order:*

1. *Change the bicarbonate* $(HCO_3)_2$ *to its equivalent monocarbonate* (CO_3) *radical.*
2. *Calcium* (Ca) *with sulphate* (SO_4).
3. *Remaining sulphate* (SO_4) *with magnesium* (Mg).
4. *Remaining sulphate* (SO_4) *with sodium* (Na).
5. *If calcium is in excess of sulphate* (SO_4), *combine calcium* (Ca) *remaining after No.* 2 *with carbonate* (CO_3).
6. *Remaining carbonate* (CO_3) *with magnesium* (Mg).
7. *Remaining magnesium* (Mg) *with chloride* (Cl).
8. *Remaining chloride* (Cl) *with sodium* (Na).
9. *Nitrate* (NO_3) *with sodium* (Na).

To illustrate the method of changing an analysis from the ionic to combined form, these rules will be applied to the following analysis which is given in terms of parts per million, p.p.m., and grains per U. S. gallon.

Example 9-2.—Analysis *A*, in ionic form.

	P.p.m.	Gr. per U. S. gal.
Calcium (Ca)	62	3.62
Magnesium (Mg)	31	1.81
Sodium (Na)	37.7	2.20
Iron oxide and alumina	3.6	0.21
Silica (SiO_2)	18	1.05
Bicarbonate $(HCO_3)_2$	250	14.59
Sulphate (SO_4)	138	8.05
Chloride (Cl)	11	0.64
Volatile and organic	56	3.27
Total solids by evaporation	480	28 00
Suspended matter	238	13.89
CO_2 (free)	16	0.93
	1,341 3	78.26

1. Change $(HCO_3)_2$ to equivalent CO_3: $250 \times {}^{60}/_{122} = 123$ p.p.m. Inspection will show that the CO_3 and SO_4 radicals exceed the Ca and Mg requirements.
2. $MgCO_3$ formed by Mg $= 31 \times \frac{84.4}{24.4} = 107$ p.p.m.
   CO_3 remaining $= 123 - (107 - 31) = 47$ p.p.m.
3. $CaCO_3$ formed by $CO_3 = 47 \times {}^{100}/_{60} = 78$ p.p.m.
   Ca remaining $= 62 - (78 - 47) = 31$ p.p.m.
4. No remaining CO_3.
5. $CaSO_4$ formed by Ca $= 31 \times {}^{136}/_{40} = 105$ p.p.m.
   SO_4 remaining $= 138 - (105 - 31) = 64$ p.p.m.
6. Na_2SO_4 formed by $SO_4 = 64 \times \frac{142}{96.06} = 94$ p.p.m.
   Na remaining $= 37.7 - (94 - 64) = 7.7$ p.p.m.
7. NaCl formed by Cl $= 11 \times \frac{58.5}{35.45} = 18$ p.p.m.

Analysis *B*: Analysis *A* reported in combined form.

	P.p.m.	Gr. per U. S. gal.
Calcium carbonate	78	4.55
Calcium sulphate	105	6.13
Magnesium carbonate	107	6.24
Silica	18	1.05
Iron oxide and alumina	4	0.23
Sodium sulphate	94	5.44
Sodium chloride	18	1.05
Volatile and organic	56	3.27
Total solids by evaporation	480	28.00
Suspended solids	238	13.89
Free carbon dioxide	16	0.93
	1,214	70.78

From the water analysis, given in the ionic form, the alkalinity or hardness characteristics can be determined from the following relations.

1. CO_3 p.p.m. $\times$ 1.667 = **carbonate alkalinity** expressed as $CaCO_3$ p.p.m.
2. HCO_3 p.p.m. $\times$ 0.819 = **bicarbonate alkalinity** expressed as $CaCO_3$ p.p.m.
3. OH p.p.m. $\times$ 2.941 = **hydroxyl alkalinity** expressed as $CaCO_3$ p.p.m.
4. Ca p.p.m. $\times$ 2.497 = **calcium hardness** expressed as $CaCO_3$ p.p.m.
5. Mg p.p.m. $\times$ 4.115 = **magnesium hardness** expressed as $CaCO_3$ p.p.m.

6. **Total alkalinity** expressed as $CaCO_3$ p.p.m. × 1.06 = **total alkalinity** expressed as Na_2CO_3 p.p.m.

7. SO_4 p.p.m. × 1.479 = **sodium sulphate** p.p.m.

The common hardness test is made on 100 c.c. of the water to be tested by dropping into the water sample 0.2 c.c., at a time, of standard soap solution and shaking the sample vigorously after each drop. As a lather will not be formed until all of the calcium and magnesium salts have reacted with the soap to form insoluble calcium and magnesium soaps, the quantity of soap required to produce a lather is an indication of the calcium and magnesium hardness. A typical reaction with soaps follows:

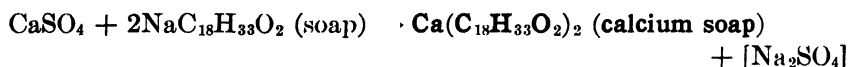
$$CaSO_4 + 2NaC_{18}H_{33}O_2 \text{ (soap)} \rightarrow \mathbf{Ca(C_{18}H_{33}O_2)_2} \text{ (calcium soap)} + [Na_2SO_4]$$

A study of reactions of calcium and magnesium salts with soap reveals the fact that the same quantity of soap combines with one mole-

TABLE 9-4.—TYPICAL WATER ANALYSES FROM DIFFERENT LOCATIONS IN THE UNITED STATES, GRAINS PER U. S. GALLON

Sample	1	2	3	4	5	6	7	8	9	10	Possible effect in boiler water
$CaSO_4$	8.12	13.00	3.37	4.90	10.78	1.11	5.64	5.18	....	19.30	Scale, corrosion
$CaCO_3$	7.89	12.30	....	3.44	9.40	1.46	6.77	3.20	13.25	3.56	Scale
$MgCO_3$	2.10	5.36	....	....	4.78	0.99	....	2.54	4.23	5.15	Scale
$MgSO_4$	....	....	2.27	....	....	....	0.86	....	2.64	....	Scale, corrosion
Silica SiO_2	0.17	0.82	1.17	1.17	0.76	0.17	0.58	0.91	0.51	0.76	Scale
Fe and Al oxides	....	....	0.17	....	....	....	0.21	0.21	0.23	0.12	
H_2SO_4	....	....	0.06								Corrosion
$MgCl_2$	....	....	....	1.92	....	....	9.23				Scale, corrosion
Total incrusting solids	18.28	31.48	9.04	11.43	25.72	3.73	23.29	12.04	20.86	28.89	
Organic	2.74	6.32	1.82	2.45	3.21	0.76	7.28	5.39	6.54	1.93	Corrosion, foaming
$NaSO_4$	....	0.64	1.05	....	1.17	0.53	....	....	1.88		
NaCl	....	2.28	6.06	0.70	1.11	0.29	54.92	1.10	1.15	0.76	Inert or corrosive
$NaNO_3$	0.47	....	....	8.10							

cule of calcium or magnesium salt. That is one molecule of $CaSO_4$ (mol. wt. 136) has the same soap combining capacity as one molecule of $CaCO_3$ (mol. wt. 100). Hence the weight of any calcium or magne-

sium salt in a water analysis can be changed to the equivalent weight of calcium carbonate, on a soap-combining-capacity basis, by multiplying by the ratio of the molecular weights. The total grains of equivalent calcium carbonate obtained by the addition of the equivalent weights of calcium carbonate hardness is a measure of the hardness of water and is termed the degrees U. S. hardness.

The multiplying factors to reduce other hardness to equivalent calcium carbonate hardness follow:

Magnesium carbonate	1 19
Magnesium bicarbonate	0 68
Magnesium sulphate	0 833
Magnesium chloride	1 05
Magnesium nitrate	0 68
Calcium carbonate	1 00
Calcium bicarbonate	0 62
Calcium sulphate	0 735
Calcium chloride	0 901
Calcium nitrate	0 61

Example 9-3.—Determine the hardness (U S degrees) of the water from sample No 1, Table 9-4

Solution.

$$CaSO_4 = 8\ 12 \times 0\ 735 = 5\ 96$$
$$CaCO_3 = 7\ 89$$
$$MgCO = 2\ 10 \times 1\ 19 = 2\ 50$$

U S degrees hardness = 16 35

171. Treatment of Water Containing Incrustants.—A common treatment of raw water is the lime and soda treatment using hydrated lime, $Ca(OH)_2$, and soda ash, Na_2CO_3. The chemical equations are shown indicating the reaction for each of the common impurities classed as incrustants. The precipitates are printed in italics, while the salts enclosed in brackets are the very soluble ones.

These soluble sodium salts enter the boiler in solution, and remain in solution as the water is evaporated. Hence gradually there will be built up in the boiler a strong concentration, and the effect will be troublesome foaming. In treatment which produces soluble sodium salts in feedwater, concentration in the boiler is prevented by blow down, either periodic (intermittent), or continuous. In the latter method the blow-down water passes through a heat exchanger and gives up a part of its heat to boiler feedwater.

$$Ca(HCO_3)_2 + Ca(OH)_2 = 2\mathit{CaCO_3} + 2H_2O \qquad (108)$$

calcium bicarbonate + lime hydroxide = calcium carbonate + water

$$Mg(HCO_3)_2 + Ca(OH)_2 = MgCO_3 + \mathit{CaCO_3} + 2H_2O \tag{109}$$

magnesium + lime = magnesium + calcium + water
bicarbonate hydroxide carbonate carbonate

$$MgCO_3 + Ca(OH)_2 = \mathit{Mg(OH)_2} + \mathit{CaCO_3} \tag{110}$$

magnesium + calcium = magnesium + calcium
carbonate hydroxide hydroxide carbonate

$$MgCl_2 + Ca(OH)_2 = \mathit{Mg(OH)_2} + [CaCl_2] \tag{111}$$

magnesium + calcium = magnesium + calcium
chloride hydroxide hydroxide chloride

$$CaCl_2 + Na_2CO_3 = \mathit{CaCO_3} + [2NaCl] \tag{112}$$

calcium + sodium = calcium + sodium
chloride carbonate carbonate chloride

$$CaSO_4 + Na_2CO_3 = \mathit{CaCO_3} + [Na_2SO_4] \tag{113}$$

calcium + sodium = calcium + sodium
sulphate carbonate carbonate sulphate

$$MgSO_4 + Ca(OH)_2 + Na_2CO_3 = \mathit{Mg(OH)_2} + \mathit{CaCO_3} + [Na_2SO_4] \tag{114}$$

magnesium sulphate + [calcium hydrodide, sodium carbonate] = magnesium hydroxide + calcium carbonate + sodium sulphate

The following example illustrates two methods of calculating the weight of reagent required to eliminate scale-forming salts. The first method is by calculating the ratio of molecular weights, reagent to incrustant. If the analysis is given in the combined form and the carbonates of magnesium and calcium are given as monocarbonates, these must be changed to the equivalent bicarbonate before calculating the weight of calcium hydroxide required. The results from the chemical equations are based on 100 per cent pure chemicals, lime hydroxide and soda ash. The second method is by using Table 9-5 (page 278) which is based on commercial chemicals.

An excess of 1½ gr. per U. S. gallon (0.21 lb. per 1,000 gal.) of sodium carbonate is recommended to lessen the solubility of the calcium and magnesium salts.

Example 9-4.—The following water is to be treated by the lime and soda process. Calculate the weight of lime hydroxide and soda ash required; both in pounds per 1,000 U. S. gal.

Water analysis, grains per U. S. gal., is as follows:

Constituent	Grains per U. S. gal.
Calcium carbonate ($CaCO_3$)	9.92
Calcium sulphate ($CaSO_4$)	7.35
Magnesium sulphate ($MgSO_4$)	6.42
Magnesium chloride ($MgCl_2$)	0.29
Sodium chloride (NaCl)	1.40
Total solids	25.38

Solution.—To soften this water both lime and soda ash must be used.

In natural untreated water at room temperature, the $CaCO_3$ will be in the bicarbonate form and must be so considered in determining the treatment. Hence the following method will be used in determining the $Ca(HCO_3)_2$ equivalent of the 9.92 gr. per gallon $CaCO_3$:

$$\frac{Ca(HCO_3)_2}{CaCO_3} = \frac{162}{100} = 1.62$$

$$1.62 \times 9.92 = 16.07 \text{ gr. per gallon } Ca(HCO_3)_2$$

Substituting in the water analysis the value 16.07 gr. per gallon $Ca(HCO_3)$ for 9.92 gr. per gallon $CaCO_3$, the calculations of the requirements of lime and soda ash shown in the table on page 276 (two methods being illustrated).

Another method of calculating the lime and soda ash requirements is to change each salt of the water analysis to its calcium carbonate equivalent. Then the summation of the total items that require $Ca(OH)_2$ multiplied by the factor from Table 9-5 gives the $Ca(OH)_2$ requirement. For soda ash the $MgSO_4$ equivalent is calculated. Using the data of the preceding example:

	Combine with $Ca(OH)_2$	Na_2CO_3
$CaCO_3$ 9.92 × 1.00 =	9.92	
$CaSO_4$ 7.35 × 0.735 =	....	5.40
$MgSO_4$ 6.42 × 0.833 =	5.34	5.34
$MgCl_2$ 0.29 × 1.05 =	0.305	0.305
NaCl 1.40		
	15.565 gr. per gal.	11.045 gr. per gal.

$$Ca(OH)_2 \text{ required} = 15.565 \times 0.116 = 1.8 \text{ lb. per 1,000 gal.}$$

$$Na_2CO_3 \text{ required} = 11.045 \times \frac{0.126}{0.833} = 1.67 \text{ lb. per 1,000 gal.}$$

$$\text{Recommended excess} = \frac{0.21}{1.88} \text{ lb. per 1,000 gal.}$$

Other reagents used in softening include caustic soda, sodium aluminate, barium salts, monosodium phosphate, disodium phosphate, trisodium phosphate. Barium salts, though more expensive than sodium carbonate, are preferable because the resulting barium sulphate is an insoluble precipitate.

A simple method, recommended by *W. J. Ryan*, for the calculation of the amount of chemicals required for water treatment follows:

$$\text{Lime} = \frac{\text{alkalinity} + \text{Mg hardness} + (2.3 \times CO_2)}{145.8}$$

$$\text{Soda ash} = \frac{\text{total hardness} - \text{alkalinity}}{113}$$

Impurity	Molecular weight: Ratio	Molecular weight: Reagent/Impurity	Weight of impurity (Gr. per gal.)	Weight of reagent (Gr. per gal.)	Weight of reagent, lb. per 1,000 gal. water
First Method—ratio of combining weights; calculation of $Ca(OH)_2$ requirements					
$Ca(HCO_3)_2$	$\frac{74}{162}$ =	0.457 ×	16.07 =	7.34 ×	$\frac{1,000}{7,000}$ = 1.05
$MgSO_4$	$\frac{74}{120}$ =	0.615 ×	6.42 =	3.94 ×	$\frac{1,000}{7,000}$ = 0.563
$MgCl_2$	$\frac{74}{95.3}$ =	0.776 ×	0.29 =	0.225 ×	$\frac{1,000}{7,000}$ = 0.032
		Total $Ca(OH)_2$, lb. per 1,000 gal.			1.645
Calculation of Na_2CO_3 requirements					
$CaSO_4$	$\frac{106}{136}$ =	0.78 ×	7.35 =	5.73 ×	$\frac{1,000}{7,000}$ = 0.82
$MgSO_4$	$\frac{106}{120}$ =	0.883 ×	6.42 =	5.67 ×	$\frac{1,000}{7,000}$ = 0.81
$MgCl_2$	$\frac{106}{95.3}$	1.112 ×	0.29 =	0.322 ×	$\frac{1,000}{7,000}$ = 0.046
		Total Na_2CO_3, lb. per 1,000 gal.			1.676
		Recommended excess			.21
					1.886
Second method—values of Table 9-5; calculation of $Ca(OH)_2$ requirements					
$CaCO_3$	.....	....	9.92	× 0.116	= 1.15
$MgSO_4$	.....	....	6.42	× 0.094	= 0.604
$MgCl_2$	.....	....	0.29	× 0.124	= 0.036
		Total $Ca(OH)_2$, lb. per 1,000 gal.			1.790
Calculation of Na_2CO_3 requirements					
$CaSO_4$	.....	....	7.35	× 0.11	= 0.809
$MgSO_4$	.....	....	6.42	× 0.126	= 0.809
$MgCl_2$	.....	....	0.29	× 0.16	= 0.046
		Total Na_2CO_3, lb. per 1,000 gal.			1.664
		Recommended excess			0.21
					1.874

$$\text{Trisodium phosphate} = \frac{\text{total hardness} \times 2.5}{120}$$

$$\text{Disodium phosphate} = \frac{\text{total hardness} \times 2.38}{120}$$

$$\text{Monosodium phosphate} = \frac{\text{total hardness} \times 0.92}{120}$$

Example 9-5.—Calculate the lime and soda ash requirements for Ex. 9-4 by the Ryan method.

Name	Gr. per gal.	P.p.m.	Ionic form	
			Radical	P.p.m.
$CaCO_3$	9.92	170	Ca	105
$CaSO_4$	7.35	126	Mg	23
$MgSO_4$	6.42	110	CO_3	102
$MgCl_2$	0.29	5	SO_4	177
NaCl	1.40	24	Cl_2	19
			Na	9
	25 38	435		
				435

Solution (see page 271).

$CaCO_3$ alkalinity = 102 × 1.667 = 170

Mg hardness = 23 × 4.115 = 94.5

$$Ca(OH)_2 = \frac{170 + 94.5}{145.8} = \frac{264.5}{145.8} = 1.81 \text{ lb. (commercial lime 90 per cent) per 1,000 gal.}$$

Mg hardness = 23 × 4.115 = 94.5

Ca hardness = 105 × 2.497 = 262.0

Total hardness 356.5

$$Na_2CO_3 = \frac{356.5 - 170}{113} = \frac{186.5}{113} = 1.65 \text{ lb. per 1,000 gal.}$$

Recommended excess = 0.21

1 86 lb. per 1,000 gal.

172. Chemical Softening Systems.—Chemical softening plants are classified as *intermittent* or *continuous*, depending on the flow of water through the softener. They are also classified as either *cold* or *hot* processes.

173. Cold-process Softening System.—In the typical *cold-process, intermittent plant*, the raw water is fed into a treating tank and the proper amount of chemicals is then added. By means of motor-driven paddle wheels the chemicals and water are thoroughly mixed.

The water is then allowed to stand quiet for a period of time, varying from 6 to 10 hrs., so the insoluble salts may precipitate and collect at the bottom of the tank. The water is taken from the tank through a float-discharge connection. Hence, the water is taken from the

TABLE 9-5.—REAGENTS REQUIRED IN LIME AND SODA PROCESS FOR TREATING 1,000 U. S. GALLONS OF WATER PER GRAIN OF CONTAINED IMPURITIES PER GALLON[1]

Name	Lime, lb.	Soda, lb.	Name	Lime, lb.	Soda, lb.
Calcium carbonate.......	0.116	.....	Ferrous carbonate..	0.169	
Calcium sulphate........	.....	0.11	Ferrous sulphate........	0.070	0.110
Calcium chloride........	.....	0.137	Ferric sulphate.........	0.074	0 126
Calcium nitrate.........	.....	0.093	Aluminum sulphate.. ..	0.087	0 147
Magnesium carbonate....	0.28	.....	Free sulphuric acid... ..	0.120	0 155
Magnesium sulphate.....	0.094	0.126	Sodium carbonate... ..	0.093	
Magnesium chloride......	0.124	0 16	Free carbon dioxide.. ..	0.223	
Magnesium nitrate	0.078	0.104	Hydrogen sulphate.....	0.288	
			Magnesium bicarbonate.	0.16	
			Calcium bicarbonate ..	0 053	

[1] Based on lime containing 90 per cent calcium oxide. Based on soda containing 58 per cent sodium oxide. (Courtesy L. M Booth Co)

surface, insuring a minimum content of precipitated solids. It is common practice to pass this water through a filter and thence to the boiler feedwater heater. Because of the time required for settling, at least two, and usually more, treating tanks are required. This arrangement assures that there is always a tank of purified water in readiness.

Intermittent plants are often used as hot-process plants; that is, for treating water that has been previously heated. As may be seen in Fig. 162, heat greatly hastens the reaction. The number of tanks thus required is considerably less. The principal advantage of the intermittent plant lies in the fact that the water can be given a very accurate dose of the chemical reagent, as the chemicals are simply weighed and added to a tank of water of known volume which has been analyzed for the amount of incrusting solids.

174. Hot-process Softening System.—The most common type in use, however, is the *hot-process, continuous softener.* This type is more flexible, handling wide fluctuations in water flow, and requires less floor space than the intermittent type. There are in use cold-process softeners of the continuous type, but on referring to Fig. 162 the superiority of the hot-process system is clearly shown. From

these curves it is seen that the elimination of magnesium hydrate in water heated to 205°F. is more rapid than in water at 50°F.

There are a number of makes of hot-process softeners, but, in general, all have the following principal parts:

1. Water heater
2. Chemical proportioner
3. Reaction and sedimentation chamber
4. Filters

The Cochrane hot-process softener (Fig. 163) shows in detail the equipment listed above. At the top of the reaction chamber is

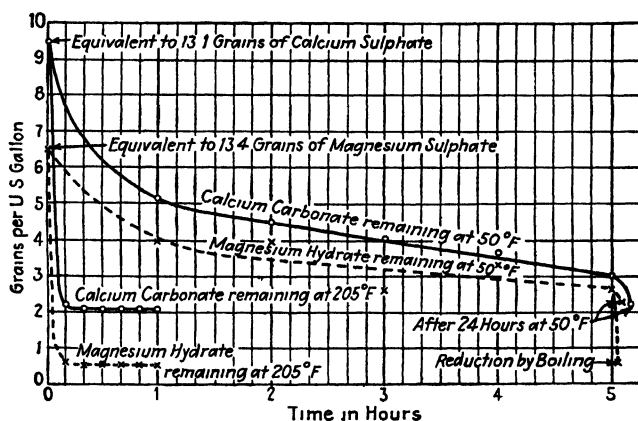



FIG. 162.—Curves showing the effect of heat on the precipitation of calcium and magnesium

an open feedwater heater, in which the raw water is heated to about 205°F. by exhaust steam. The heater is equipped with an oil separator and a vent for expelling gases. From the heater trays, the heated water falls into the large reaction or settling tank in which the water level is kept constant and near the top by a float-operated valve in the raw-water supply line.

The chemical-solution tank will hold a 12-hr. supply of chemical solution, consisting of lime and soda ash. A propeller agitator driven by a belt is used to keep the chemicals, particularly the lime, in suspension. The circulating pump, as shown, takes the chemical solution from near the bottom of the tank and delivers it to the chemical proportioner considerably in excess of the amount required by the softener. The proportioner delivers to the feed pump just the amount required, depending on the flow of raw water, and allows the excess chemical solution to drop back into the chemical solution tank.

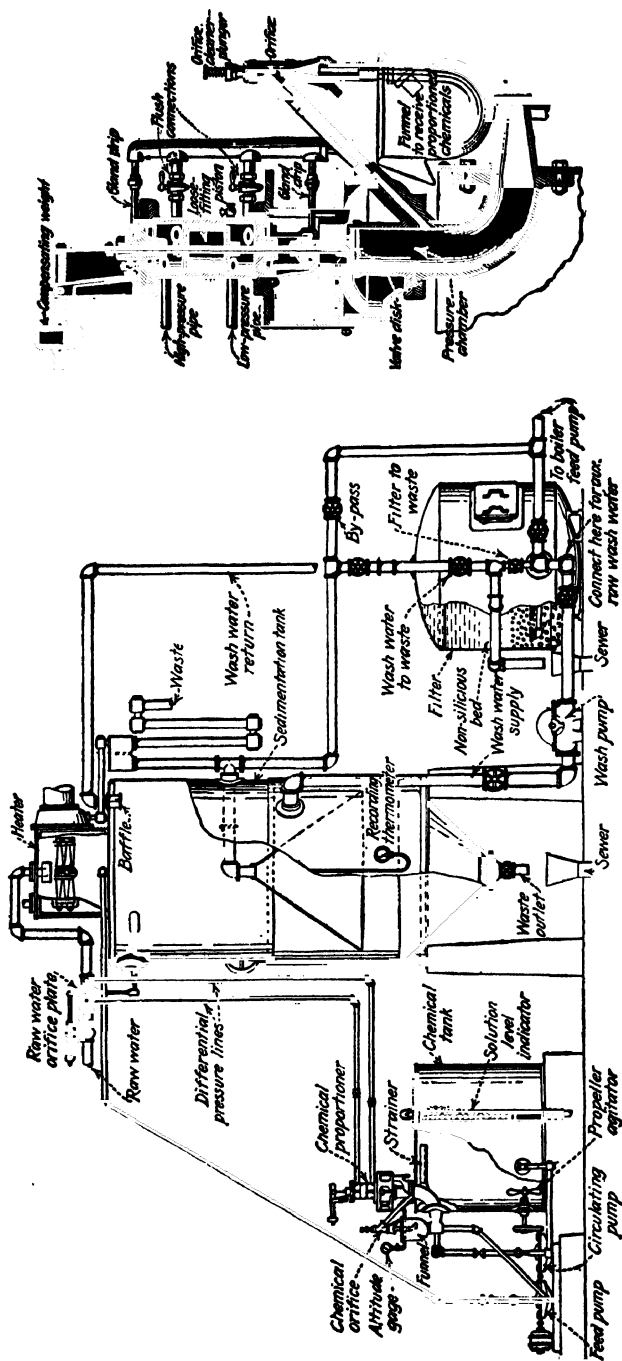



FIG. 163.—Cochrane hot-process continuous softener.

An orifice plate is placed in the raw-water line between the regulating valve and heater. When a flow of water occurs, the pressure on the approach side of this orifice is greater than on the outlet side. Small pipes connect the two sides of this orifice to the cylinder of the proportioner (see insert in Fig. 163), the approach side to the top and the outlet side to the bottom of the cylinder.

In the cylinder is a piston of light material, with a piston rod extending through both ends of the cylinder. At the upper end the piston rod is connected, through a flexible suspension link, to a lever which carries a weight to balance the weight of the piston and other parts. At the bottom end, the piston rod is attached to a valve disc of the same area as the piston. Hence, the pressure on the chemical discharge from the circulating pump is always equal to the difference in pressure at the raw-water orifice.

Consider the operation of this proportioner when there is no flow of water. The pressure differential is zero and, due to the balance weight, the piston is retained at the top of the cylinder, and the valve disc far off its seat. In this position there is no resistance to the flow of chemical solution through the main outlet, and all of the solution drops back to the solution tank. But, with water flowing through the raw-water orifice, the pressure difference is transmitted to the cylinder. The piston then moves down, and the valve disc imposes a pressure on the chemical solution in the pressure chamber equal to the difference in pressure on the two sides of the orifice. Because of this pressure a part of the chemicals will be forced up the inclined pipe and will fall, by gravity, through the chemical orifice to the chemical feed pump and be discharged to the top of the settling tank. Hence, the flow of chemicals is proportioned exactly to the flow of water.

In the settling tank the water gravitates slowly downward, as the insoluble precipitates are formed and drop to the bottom. From here they can be removed by opening the waste or blow-off valve. The treated water is removed through an inverted funnel. It passes through a filter and is then ready for the boiler.

Another type of softener, using soda ash, caustic soda, or both, is shown in Fig. 164. A plunger pump (not suitable for lime) is used for pumping the chemicals and is attached to the boiler feed pump or to a pump for feeding raw water to the sedimentation tank. Thus, the chemical feed is directly proportional to the water feed. Other parts of this softener plant are as previously described.

An important part of most water-softening systems is the filter. A filter, as shown in Fig. 163, consists, in general, of a tank partially

filled with granular materials. The treated water is passed through this material, thereby removing the suspended solids. Filtering materials used are sand, gravel, coke, stones, charcoal, shavings, etc.

175. Water Filters.—Filters commonly used for boiler water have a layer of gravel or coarse stone at the bottom, from 10 to 20 in. in thickness. Over this is placed a layer of sand or other fine material, to an average thickness of 30 in. During filtration, water enters at the top and leaves the filter at the bottom. In boiler-plant practice water passes through the filter at comparatively high rates (2 to 5 gal. per square foot of filter area per minute). Filters may consist of an

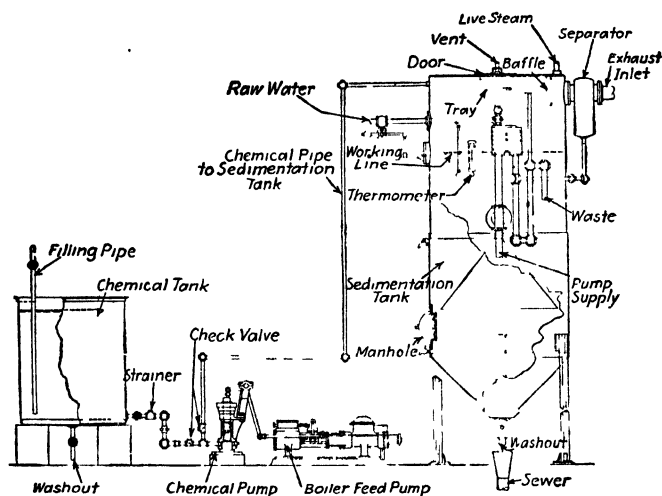



FIG. 164.—Cochrane hot-process softener with plunger chemical pump.

open tank at atmospheric pressure or a closed tank through which water can be forced under pressure. When the surface of the filtering material becomes coated with mud, it is cleaned by reversing the flow of water through the filter and washing the filter material. During washing, the material is agitated by air jets or by the high velocity of the wash water.

Filtering is often done when chemical treatment is unnecessary and is warranted if the feedwater contains any appreciable amount of suspended solids, grease or sewage. Such material introduced in the boiler will cause foaming and priming, and possibly damaged tubes.

As a preliminary step to filtration, coagulation is sometimes used to help clarify water carrying a large amount of suspended solids. There are many chemicals, which, added to water, will cause coagulation. The one commonly used is aluminum sulphate (filter alum).

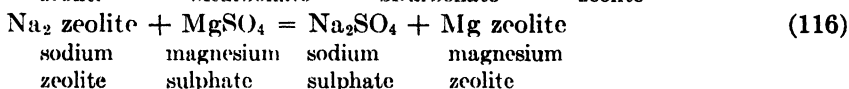
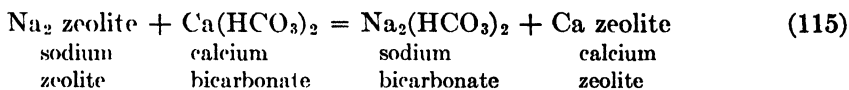
When alum is added to alkaline water, gelatinous substances are formed, and this action is called coagulation. These gelatinous substances build up with each other and with entrained suspended solids until relatively large bodies of spongelike material are formed.

When coagulated water passes through sand filters the sand catches these gelatinous bodies, and they form a spongelike surface on the sand, through which the water can pass, but which entrains suspended solids.

Care must be taken in determining the amount of coagulant required, as an excess may increase the permanent hardness or introduce corrosive properties to the water. Certain finely divided matter, such as clays, hinder coagulation, and certain acids also retard this action. Sufficient time must be allowed for thorough mixing of the water and coagulant. Ordinarily, less than one hour will suffice. This, however, depends upon the condition of the water.

176. Zeolite Softeners.—Zeolites are either natural or artificial silicates of sodium and aluminum. Water passing through zeolite beds lose calcium and magnesium, exchanging the same for sodium. Zeolites are known as base-exchange chemicals. The following equations give the chemical reactions for the removal of one salt of calcium and one of magnesium by the base-exchange process. The equations for the removal of other scale-forming materials are similar.

$$\underset{\text{sodium zeolite}}{Na_2 \text{ zeolite}} + \underset{\text{calcium bicarbonate}}{Ca(HCO_3)_2} = \underset{\text{sodium bicarbonate}}{Na_2(HCO_3)_2} + \underset{\text{calcium zeolite}}{Ca \text{ zeolite}} \qquad (115)$$

$$\underset{\text{sodium zeolite}}{Na_2 \text{ zeolite}} + \underset{\text{magnesium sulphate}}{MgSO_4} = \underset{\text{sodium sulphate}}{Na_2SO_4} + \underset{\text{magnesium zeolite}}{Mg \text{ zeolite}} \qquad (116)$$

After the sodium has been exhausted, the zeolite may be recharged by passing a solution of sodium chloride (common salt) through it. The zeolite acquires the sodium in exchange for the calcium and magnesium, and a solution of calcium chloride and magnesium chloride is discharged to the drain.

The zeolite process gives water with zero hardness. However, if the raw water contains a large amount of carbonates (temporary hardness), the treated water will contain a large amount of soluble sodium salts which may cause foaming or result in an alkaline water. Often water, high in carbonates, is passed through a continuous lime treatment and filtered before going to the zeolite tank.

The base exchange power between regenerations refers to the total number of grains of hardness, expressed as calcium carbonate, which

1 cu. ft. of zeolite exchanges up to the point when water of zero hardness is no longer produced. With non-porous zeolite of a grain size of about ½ mm., the softening capacity per cubic foot is about 2,500 to 3,000 gr., as $CaCO_3$. With porous zeolite the capacity per cubic foot increases to about 10,000 gr. as $CaCO_3$.

In comparison with other chemical softeners, the zeolite process produces softened water with 1 to 1½ gr. of incrusting solids per U. S. gallon; hot-process lime and soda-ash softeners, 1 to 3 gr.; and the cold process lime and soda softeners, from 3 to 5 gr.

177. Boiler Compounds.—Boiler compounds are extensively used for scale prevention, especially in smaller sized plants. It is generally agreed that external chemical treatment is preferable, but if this is impossible, the use of a good boiler compound is better than using a poor feedwater with no treatment. For such doctoring of boiler water, an accurate and frequent chemical analysis of the water is necessary. A boiler compound suitable for one water may be useless with another. Reputable manufacturers are able to prepare compounds that will be useful in softening a particular water.

It should be emphasized that internal treatment does not eliminate the impurities from the water, it simply changes them to different combinations. Usually, the results sought are: first, to precipitate materials as a soft sludge which can be blown down from the boiler; second, to eliminate any salts that have corrosive action. Sodium hydroxide, sodium silicate, sodium phosphate and other sodium compounds form the principal active ingredients of boiler compounds.

Graphite is often used to prevent scale, but its use is questionable as it may have highly corrosive action. Its tendency is to attack the metal under the scale and in this manner break it loose.

Antifoaming compounds are used with waters high in soluble sodium and potassium salts. Castor oil has a marked inhibiting effect on the foaming of water and is the principal ingredient of these compounds.

178. Steam Traps.—Steam traps are used in power plants to automatically remove condensed steam from steam lines, steam heating apparatus and other equipment. They are designed to discharge the water to a receiver and to reduce to a minimum any loss of steam. Boiler *return traps* are placed above the boiler and are subjected to the boiler pressure during the periods of discharge. *Non-return traps* discharge condensate to a hot well or other receiver in which the pressure is usually atmospheric.

Traps commonly used for power-plant service are included in the following classes:

1. Float.
2. Bucket.
3. Tilting.
4. Expansion.

A *float trap* is shown in Fig. 165. The body of the trap is constructed of cast iron. The ball float is seamless and is made of copper,

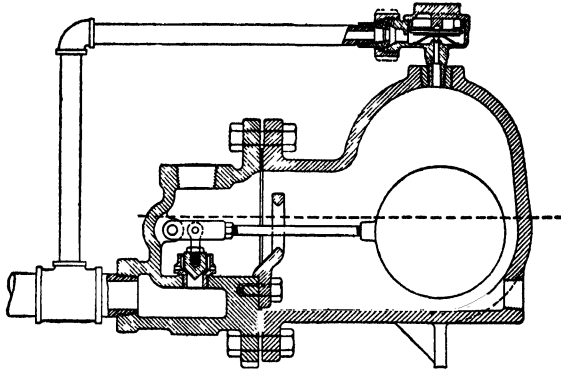



FIG. 165.—McAlear float trap.

and the valve parts are of bronze. The float is attached to one end of a lever which is pivoted to the body at the other. The discharge valve is connected to the lever, near the pivot. Steam and condensate

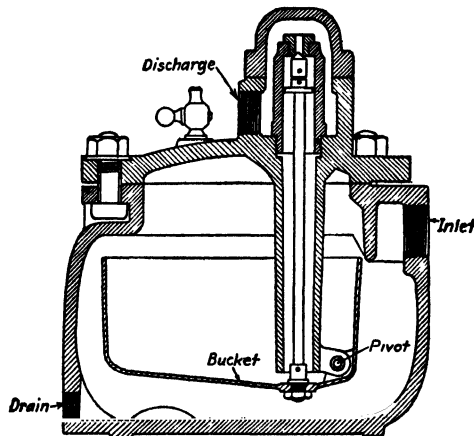



FIG. 166.—Cochrane bucket trap.

enter the trap at the inlet, and when sufficient water has collected to raise the float, the discharge valve is opened and the steam pressure, acting on the surface of the water, causes it to flow out at rather high velocity. As the water level drops the discharge valve is closed. The

action may be either intermittent or continuous. The trap, as shown, is equipped with a thermostatic valve to remove the air which collects in the top of the float chamber. This valve is automatically vented to the discharge line, and it is used for low and medium pressures only.

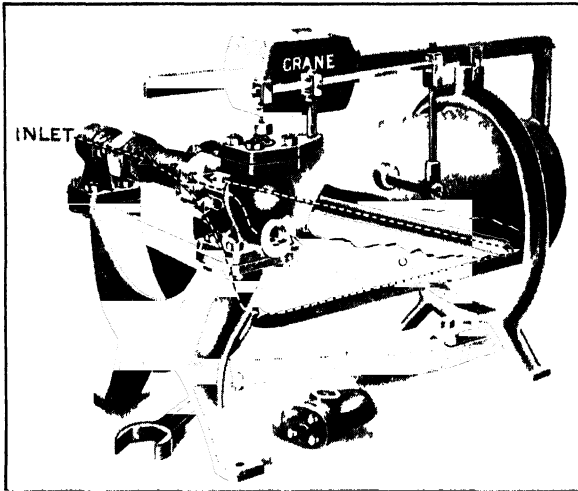



FIG. 167.—"Crantilt" trap.

With pressures of 150 lb. or over a hand-operated pet cock is provided for this purpose.

A typical *bucket trap* is shown in Fig. 166. Water fills the space between the bucket and the walls of the trap, and the empty bucket, floating on the water, closes the discharge valve. When the water rises above the edge the bucket is filled, causing it to drop, opening the discharge valve and discharging the water. When empty, the bucket rises, closing the valve, and the operation of the trap is repeated.

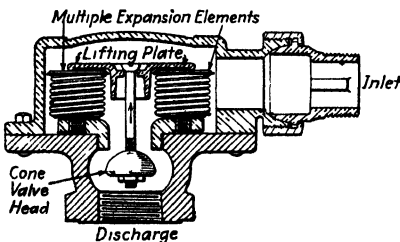



FIG. 168.—"Sarco" expansion trap.

A typical *tilting trap* (Fig. 167) has an inclined receiving tank, balanced to incline in one position when empty. The trunnions, about which the tank pivots, provide the inlet and discharge passages. In the empty position, the discharge valve is closed. When the tank fills, its weight overcomes the force of the counterbalance, and the tank rotates to a horizontal position. This opens the discharge, and the water is emitted by the pressure of the steam above it. The action, thus, is intermittent.

The *expansion trap* (Fig. 168) operates on the expansion principle. The bellows is of seamless, flexible metal, and contains a volatile fluid. In contact with steam, the fluid vaporizes, and the resulting pressure expands the bellows and closes the valve. In contact with cold water, a contraction occurs, which opens the valve, discharging the condensate.

Problems

1. A closed, single-flow water heater is to be installed to heat water for four boilers rated at 500 hp. each. The maximum load on each boiler is 300 per cent of the rating, with steam pressure of 275 lb. per square inch absolute, and quality of 99.3 per cent. The temperature of the steam to the heater is 250.5°F.; the water enters the heater at 120°F. and leaves at 225°F. Calculate the theoretical heating surface required if the heater is to supply water to three of the boilers at their maximum rating.

2. What heating surface would be required in Problem 1, if a multiple-flow heater with corrugated copper tubes is used?

3. A 2,500 sq. ft., coil-pipe heater with iron tubes heats water from 80 to 170°F. The steam temperature is 328°F. How much water is being heated?

4. In a small power plant, a boiler developing 680 hp. receives water at 80°F., evaporating steam at 120 lb. per square inch absolute, and a quality of 0.986. Calculate the percentage increase in the actual evaporation if the feedwater temperature is increased to 250°F., assuming no change in the heat absorption.

5. Two samples of water give pH numbers of pH3 and pH12, respectively. Which sample will give an acid reaction and which alkaline? What is the concentration of hydrogen ions in each? What reagent would be used to neutralize each water?

6. Determine the hardness, U. S. degrees, of the 10 samples of water in Table 9-4.

7. Using the chemical equations, calculate the weight (pounds per 1,000 gal.) of calcium hydroxide and of soda ash required for the water analyses of Table 9-4. Consider only the carbonates of calcium and magnesium and the sulphates of calcium and magnesium.

8. The same as Problem 7, using the data given in Table 9-5 (page 278).

9. Test data: coal, pounds per hour 25,500; ash and refuse, pounds per hour 1,655; carbon in ash and refuse, per cent, 20; evaporation, 8 lb. of water per pound of coal. Raw water analysis, grains per gallon: $CaSO_4$ 4.95, $CaCO_3$ 8.65, $MgCO_3$ 3.05, $MgSO_4$ 2.42, $CaCl_2$ 1.95. Calculate the calcium hydroxide required for treatment, pounds per hour, if 20 per cent of evaporation is raw water.

10. Boiler test: Hourly quantities: steam 175,400 lb., coal, as fired, 16,500 lb., ash and refuse, 1,650 lb. The boiler uses 15 per cent raw make-up water, analysis in grains per gallon: $CaSO_4$ 3.42, $CaCO_3$ 8.95, $MgCO_3$ 5.05, $MgSO_4$ 5.95. Calculate pounds per hour (*a*) calcium hydroxide; (*b*) soda ash required.

11. Boiler test data: coal (as fired): ultimate analysis, per cent, S 1.05, H 5.12, C 70.39, N 1.55, O 12.55, A 9.34; coal per hour, 6,500 lb.; ash and refuse per hour, 760 lb.; C in ash and refuse, per cent, 20; flue-gas analysis, per cent, CO_2 13.2, O_2 6.2, CO 0.4; temperatures, outside air, 45°F., entering chimney 560°F., average in chimney, 470°F.; boiler room, 82°F.; barometer, 29.4 in. Hg; gas velocity in chimney, 20 ft. per second; area of air duct from fan, 12 sq. ft.;

water evaporated, 49,000 lb. per hour; water temperatures: open F.W. heater, in, 80°F., out, 195°F.; closed F.W. heater, in, 190°F., out, 270°F.

NOTE:—The water equipment consists of one open, and one closed F.W. heater, and one economizer for each boiler, with chemical equipment for make-up water.

a. In the open F.W. heater, the heat is supplied by 90 per cent dry steam at 18 lb. abs. How much steam is required per hour?

b. The closed, single-flow, brass-tube heater was designed for a heating capacity just double the present rate of water flow. The heat medium is steam, 95 per cent dry, at 50 lb. abs., the condensate at 281°F. removed by a trap. What is the surface of the heater?

c. In the closed heater of part *b*, how much steam per hour is being used, at the rate of water flow given in the test data?

d. Raw-water analysis, grains per gallon: $CaSO_4$ 8.52, $CaCO_3$ 12.32, $MgCO_3$ 4.78, $MgSO_4$ 2.64, $MgCl_2$ 1.95. The make-up water is treated at 190°F. and amounts to 15 per cent of the total boiler evaporation. Calculate weight of lime hydroxide required, pounds per hour.

e. With the make-up water data as given in part *d*, calculate weight of soda ash required, pounds per hour.

CHAPTER X

STEAM-ENGINE STUDY

179. Introductory.—The steam engine receives steam, at a pressure above that of the atmosphere, and, in the cylinder, expands it to a lower pressure. During this process a part of the heat energy of the steam is transformed into mechanical energy or useful work which is delivered, through a piston, piston rod and crank, to a rotating shaft. If a generator is attached to the engine shaft, the work, in its final form, is electrical energy. Numerous other examples may be cited in the direct application of the steam engine to practically all industrial uses, both portable and stationary.

After the expansion to lower pressure, the steam with reduced heat energy is discharged from the cylinder. The heat thus discharged is then lost, at least as far as engine power is concerned. In the usual engine, only about one-tenth of the heat of the steam is transformed into useful work. This ratio varies with the initial pressure and temperature of the steam used, and the friction of the moving parts.

180. Historical Development.—The origin and development of the steam engine were due largely to necessity. During the eighteenth century all manufacturing in England was done literally "by hand," and horses furnished the power not supplied by man. Charcoal had for a long time been used in the smelting and forging of metals and, as a result, the forests became depleted. The king, then, by royal decree, prohibited the cutting of trees for the purpose of making charcoal. This brought on a serious problem for English industry. Though there were great coal reserves in England, the coal was down deep and under water. The horse-driven pumps used in the few mines were inadequate.

In 1698 Savery, a Cornish mine owner, had patented a steam-actuated pump for raising water from these mines. Savery's pump was similar in principle and design to the present pulsometer pump used to draw water, mud and gravel from excavations. A number of the pumps were placed in service. Their steam consumption, however, was enormous, and the height to which the water could be lifted was limited.

In 1711 Thomas Newcomen introduced a practical cylinder and piston steam engine with overhead walking beam construction. Steam

was admitted to the bottom of a vertical cylinder, below the piston. *The piston was then raised, partially by the action of the steam entering* the cylinder and partially by a counterweight on the walking beam. The steam valve was closed, and a jet of cold water sprayed into the cylinder, condensing the steam. With atmospheric pressure above and a vacuum below, the piston moved downward, doing work on the pump rod attached to the opposite end of the walking beam.

The steam valve of Newcomen's engine was originally operated by hand, but the story generally credited is that Humphrey Potter, the boy hired to operate the valves, "rigged up" suitable cords and catches to the overhead beam in a way so as to make the operation of the valves automatic. A number of large engines using this contrivance were later built and installed.

James Watt, an instrument maker of Glasgow, was, in 1763, repairing a model of the Newcomen engine. Being impressed by the loss of heat due to the alternate heating and cooling of the cylinder, he became interested in the elimination of this loss and built an experimental engine embodying his own ideas.

Watt's engine had a condenser separate from the engine cylinder but connected to it. The condenser was cooled by water pouring over the outside surface and also a water spray on the inside. To keep the engine cylinder hot, it was lagged with wood, and in some cases steam jacketing was used. With a top placed on the cylinder it became double acting. For improving the condenser vacuum and removing the condensed steam, an air pump was added.

Watt invented the steam-engine indicator and made very complete tests on his engine installations. His other inventions included an automatic governor, throttle valve, mercury steam-pressure gage, and glass water column on the boiler. However, his principal claim for fame lies in the fact that he discovered the large condensation loss in the Newcomen engine and was able to provide the remedy. The date of James Watt's separate condenser marked the beginning of the present steam-power age.

181. Steam-engine Classification.—There are many different features on which to base a classification of the various designs of modern steam engines. Their main points of distinction take into consideration the following:

1. The course of steam flow within the cylinder.
2. Type of valve gear.
3. Number of cylinders in which steam expands.
4. Position of cylinder axis.
5. Speed.

6. Exhaust pressure
7. Service.
8. Ratio of stroke to diameter of cylinder.

1. The first division distinguishes an engine as being of the *counter-flow*, the *parallel-flow*, or the *straight-flow* type. Synonymous names more often used to effect the same distinction are *simple, Corliss* and *unaflow* (or *uniflow*), respectively.

2. Types of valves used are *slide valve, Corliss valve,* and *poppet valve.* Slide valves are called D, piston, balanced, or multiported, depending on the design. Corliss valves, referring to the valve gear, are of either the releasing or non-releasing type. Poppet valves have no definite subnames.

3. To steam engines may be attributed the terms *single expansion, compound,* etc., according to the number of cylinders in which the steam expands. Distinct from these are the terms *single cylinder, duplex, triplex,* etc., which refer to the number of like cylinders driving one shaft.

4. The terms *horizontal, vertical* and *angle* are used, according to the position of the cylinders.

5. An engine may be classified as *high speed, medium speed,* or *slow speed.* The shaft speed ranges are, in general, 300 r.p.m. or more, 150 to 300 r.p.m. and less than 150 r.p.m., respectively.

6. *Condensing* and *non-condensing* generally refer to the exhaust pressure, and depend on whether it is above or below the pressure of the atmosphere.

7. To steam engines are attributed the names *stationary, portable, locomotive, marine* and *hoisting,* depending on the type of service for which they are designed.

8. The stroke is *long* or *short,* according to whether it is greater or less than the diameter of the cylinder.

182. The Simple Steam Engine.—An engine of this type is of the most simple design and generally consists of a single cylinder, the piston of which is connected by a piston rod, crosshead and connecting rod to a crank shaft which carries a single flywheel. The valve may be of the D or piston, slide type, operated by an eccentric on the crank shaft, and the cylinder axis may be in any position. Engines of this type are suitable for service of almost every kind, where a small amount of power is required. Their speed is generally slow, and they are seldom built in sizes of over 50 hp. An engine of this simple, slide-valve type is illustrated in Fig. 169. This figure, being a cut-away view, exposes all of the main parts.

The stationary parts of a simple steam engine include the base, frame, cylinder and valve chest, piston stuffing box, and shaft bear-

ings. The rotating parts consist of the crank shaft with the crank arm, eccentric and flywheel. The governor, if shaft governed, is usually mounted on the flywheel and rotates around the shaft, controlling the position of the eccentric.

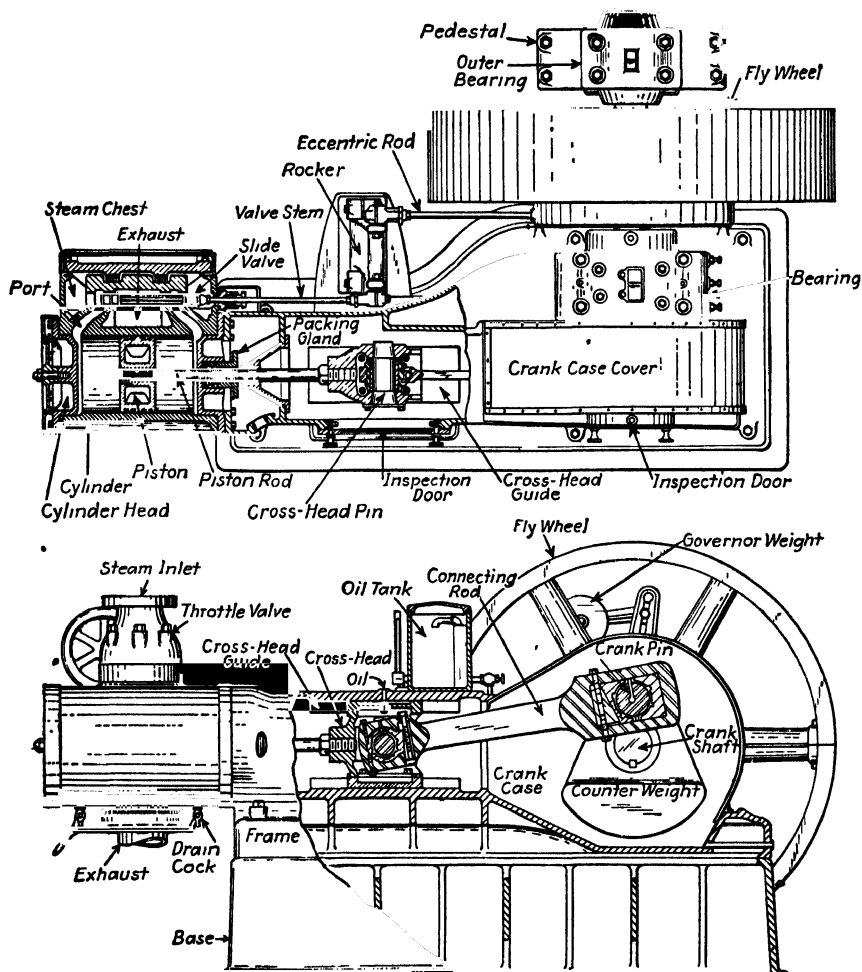



FIG. 169.—Ames side-crank, simple, slide-valve steam engine.

The reciprocating parts consist of two separate mechanisms: first, the piston, piston rod, crosshead and connecting rod, by means of which the motion of the piston in the cylinder is changed into rotative motion of the shaft; secondly, the valve gear, consisting of an eccentric disc and strap or an eccentric pin, eccentric rod, rocker arm

or valve slide, valve stem and valve. The valve gear properly regulates the flow of steam to and from the cylinder.

In operation, steam, from the steam chest, is admitted to the cylinder, near the time, in the operating cycle, when the piston is in a position to start its working stroke. The piston, being free to move, recedes, as an effect of the steam pressure, but the flow to the cylinder continues for only a portion of the stroke; that is, to the point of *cut-off*. After this, further production of power results from expansion of the steam. Expansion takes place until the piston reaches the point of release, at which point *exhaust* begins. Continuing the cycle, the piston reaches the end of its working stroke, and then begins the exhaust or return stroke. The continuous decrease in cylinder volume causes the spent steam to exhaust. This occurs until near the end of the return stroke, where *compression* begins. At this point the exhaust is stopped, and the remaining volume of steam in the cylinder is compressed by the piston during the remainder of its return travel. The effect of compression is to furnish a cushion for absorbing the portion of the kinetic energy of the piston, as it is stopped at the end of its stroke. The piston, having completed its return stroke, begins another cycle, a repetition of above-mentioned events.

An engine is *double acting* if the steam acts, alternately, on either side of the piston. This gives a power stroke during the travel of the piston in each direction, which results in practically twice the power that would be obtained if the engine were single acting.

The power given up by the steam is transmitted through the piston, piston rod, crosshead, connecting rod and crank to the engine shaft. The flywheel acts as a surge reservoir for the energy, to give smooth-running action to the moving parts and a continuous supply of power for useful work.

All events of the cycle are controlled by the valve, and its motion, relative to that of the piston, is such that these events occur at the proper time and in the proper sequence. The length of travel (unless engine is equipped with a shaft valve governor) is determined by construction, but its motion relative to that of the piston depends on the angular position of the valve eccentric with respect to the engine crank.

183. General Steam-engine Nomenclature.—Many of the terms commonly used in steam-engine practice are as follows:

Crank end refers to the end of the cylinder nearest the crank shaft; and *head end* refers to the opposite end.

Stroke refers to the linear distance traveled by the piston from one dead center to the other; the forward stroke being toward the crank, and the back or return stroke in the opposite direction.

Dead center is the position of the piston and crank when the piston is at the end of the stroke; either *H.E.* or *C.E.* When the crank is on dead center, the crank arm, connecting rod and piston rod are in a straight line.

Piston displacement is the volume, in cubic feet, swept by the piston during one stroke, and is equal to the length of stroke times the net piston area. The area of the piston rod is subtracted from the total piston area for the crank end; and also for the head end if a floating piston and tail-rod construction is used.

Volumetric clearance is the volume filled with steam when the piston is on dead center. This includes the space between the piston and cylinder head, volume of steam ports (to the valve), indicator ports, and is commonly expressed as a percentage of the piston displacement

The *eccentric disc,* or *eccentric pin,* acts as a small crank and is used to give a reciprocating motion either to the valve, direct, or its operating rod (eccentric rod). The "throw" of the eccentric, or eccentricity, is the radius of the eccentric crank, or the distance between the center of the crank shaft and the center of the eccentric. An eccentric pin has less friction than an eccentric disc.

The *slide valve* has certain terms that apply only to a valve of this type. Figure 170 shows a *D slide valve* on mid-position, in which position the eccentric crank is vertical. This is an external valve, built to have the high-pressure steam on the outside and the connection to the exhaust pipe on the inside.

Valve displacement refers to the distance of valve movement to the right or to the left of the mid-position. Maximum valve displacement indicates that the valve has moved to one of its extreme positions and the eccentric is on dead center. The *valve travel* is the total distance moved by the valve from one extreme position to the other.

Valve lap is measured when the valve is on mid-position, and is the distance, in inches, that the edge of the valve projects over the edge of the port of the cylinder. For the external valve in Fig. 170, *AB* is the *steam lap* and *CD* is the *exhaust lap* for the head end. If a valve is on mid-position and the port is open, the lap is termed negative or *clearance.* Valves are sometimes set with negative exhaust lap.

Lead is the distance, in inches, the port is open, when the engine piston is on dead center.

The four events of the steam-engine cycle are:

1. *Admission:* At this point the valve has moved off its mid-position by a distance equal to the steam lap, and is just uncovering the cylinder port, admitting high-pressure steam.

2. *Cut-off:* Here, the valve has opened the port, reached the extreme position, reversed its motion, and is just closing the port, stopping the flow of steam into the cylinder.

3. *Release:* At this point, the steam in the cylinder has expanded to near the end of the stroke. The valve has moved, from mid-position, a distance equal to the exhaust lap, and the port is being opened to permit discharge of the steam to the exhaust pipe.

4. *Compression:* This occurs when the valve, on its return, is closing the port, stopping the flow of exhaust steam from the cylinder.

184. The Principle of the Zeuner Valve Diagram.—The Zeuner valve diagram is drawn for the purpose of studying the relations between the positions of the valve and piston of a slide-valve engine. It is widely used and frequently accepted in preference to other diagrams drawn for the same purpose. For any crank position, the

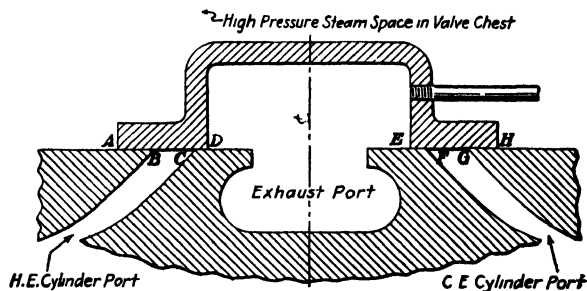



FIG. 170.—Diagrammatic drawing of a *D* slide valve. (Mid-position.)

Zeuner diagram shows the valve displacement from its mid-position. Therefore the crank may be located for each of the four events of the steam-engine cycle. A faulty valve setting becomes readily evident and suitable corrections can be determined by drawing the diagram for any particular case.

In Fig. 171 *a* the diagram indicates the relative positions of the crank and the eccentric. The engine cylinder is assumed to be horizontal, and the top of the flywheel turns away from the cylinder; *i.e.*, runs over. The circles representing the path of the crank pin and of the center of the eccentric are drawn to the same scale. The diameters *ST* and *NM* represent the length of valve travel and stroke of the piston, respectively. The angle between the position of the crank *ON* and the eccentric *OE* is 90 deg. plus an angle represented by the symbol α, and the eccentric is shown leading the crank. Angle α is called the angle of advance.

The principle of the Zeuner diagram may be understood by reference to Fig. 171 *b*. The eccentric circle is drawn to a certain scale, such

as will make the diameter MN a multiple of the valve travel. Then, the crank circle is drawn to coincide with the eccentric circle, and is necessarily drawn to a different scale. When the crank position is OC_1, the eccentric position, by using the angle 90 deg. + α, is located at OB, and the valve displacement to the right of mid-position is OV_1, to the same scale used for the eccentric circle. With O as the center and OV_1 as the radius, an arc is drawn, intersecting OB at

(*a*) Showing crank and eccentric relation. (*b*) Showing construction of Zeuner diagram.

(*c*) Zeuner diagram for head end. (*d*) Zeuner diagram for crank end.

FIG. 171.—Valve diagrams.

point 1. This process may be repeated with another crank position OC_2 and the corresponding eccentric position OE_2, giving a second point 2. If a circle is drawn with the line OM, the radius of the crank and eccentric circle, as its horizontal diameter, it will be found that all points such as 1, 2, etc., will be located on its circumference. This circle is termed the *valve circle*. If any eccentric position is drawn to the right of the vertical diameter YY', the distance from the center O to the intersection of the eccentric radius and the valve circle will, to scale, be equal to the displacement of the valve to the right of its mid-position. Similarly, the left valve circle can be drawn, the inter-

cepts of this circle on the eccentric radii representing, to scale, the *valve displacements* to the left of mid-position.

The next step is to consider the effect of swinging the eccentric through an angle of 90 deg. + α, in the direction opposite to that of its forward rotation. In the new diagram the eccentric is assumed to coincide with the crank. This shifts the line MN and the right and left valve circles back through the same angle. Then, to determine the valve displacement for any crank position, it is only necessary to draw the crank position. The valve displacement is measured by the intercept of the valve circle on the crank radius.

185. Construction of the Zeuner Valve Diagram.—The completed, head-end Zeuner diagram is shown in Fig. 171 *c*. This diagram represents the valve analysis of a double-acting, slide-valve engine. The line of centers has been moved back, so that the radius OM is at an angle α before the vertical radius OY. With O as the center, and OS_H (head-end steam lap) as the radius, an arc is drawn, intersecting the right valve circle at G and H. Crank positions OA_H and OC_H are then drawn through these points, and these positions give the location of the crank at admission and cut-off. Each of these crank positions has a length of intercept equal to the head-end steam lap. Therefore, for each position, the valve displacement to the right of mid-position is equal to the head-end steam lap. Reference to Fig. 170 will show that when the valve is displaced to the right of its mid-position by a distance equal to the head-end steam lap, the edge of the valve is over the edge of the port, in the position for admission or cut-off, depending on the direction of travel of the valve.

The crank positions for release and compression are determined by the same method. By drawing an arc with radius equal to the exhaust lap, to scale, and intersecting the left valve circle at two points, the crank positions OR_H and OK_H, drawn through these points, represent the location of the crank at release and compression for the head end. This results from the fact that the length of the intercept in each case is equal to the exhaust lap.

The following facts are useful in the construction of the Zeuner diagram:

1. The lines A_HC_H and K_HR_H (A_CC_C and K_CR_C on the crank-end diagram) are parallel to each other and perpendicular to MN.

2. The lines A_HC_H and A_CC_C are tangent to the steam-lap arc at S_H and S_C, respectively.

3. The lines K_HR_H and K_CR_C are tangent to the exhaust-lap arc at E_H and E_C, respectively.

4. If the exhaust lap is negative (the port being open to the exhaust when the valve is on its mid-position), the exhaust lap arc is on the same side of the center O as the steam lap, and intersects the same valve circle.

5. Lead is represented on the diagram by the line XL, drawn from the head-end dead-center point, and perpendicular to the line A_HC_H for the head end. For the crank end, the lead is represented by the line $X'L$, drawn from the crank-end dead center, perpendicular to the line A_CC_C.

6. If a line is drawn from M (N on the crank-end diagram), perpendicular to the axis XX', the resulting distance OB is the sum of the steam lap and the lead. Hence, the lead is represented on the diagram by the line $S'B$.

7. For convenience, the *C.E.* Zeuner is usually superimposed on the *H.E.* Zeuner, using the same crank and eccentric circle, and the same line MN.

8. The following relations are true for a slide-valve engine, no matter how the valve steam length is changed:

(Steam lap) *H.E.* + (steam lap) *C.E.* = constant
(Steam lap + exhaust lap) *H.E.* =
(steam lap + exhaust lap) *C.E.*

9. A perpendicular line from M, to crank positions OA or OC, strikes the intersection of the steam-lap arc with the crank position. The same is true of the exhaust lap and the crank positions for the exhaust events. This is useful in determining laps.

The items of data which are shown by the complete Zeuner diagram are as follows (x indicates unknown data):

		Head End	Crank End
1.*	Valve travel	for both	
2.	Angle of advance	for both	
3.	Ratio of crank to connecting rod, R/L	for both	
4.*	Exhaust lap	x	x
5.*	Steam lap	x	x
6.*	Lead, opening of the steam port at dead center	x	x
7.*	Maximum port opening	x	x

The following events are expressed as percentage of the stroke:

8.	Admission	x	x
9.	Cut-off	x	x
10.	Release	x	x
11.	Compression	x	x

The items marked * are to the same scale used for the eccentric circle.

In general, if item No. 3 and three other items for one end of the cylinder, and two items for the other end are known, the Zeuner diagram can be drawn, and all the above data can be determined.

186. Application of the Zeuner Valve Diagram.—At O draw a diameter PP', perpendicular to MN. Starting with the crank position OP, observe the valve action as the crank turns one revolution about the shaft. The crank position OP is tangent to both valve circles, and as there is no intercept by either valve circle, the valve displacement from mid-position is zero. In other words, when the crank is at OP, as shown in Fig. 171 c, the valve is on its mid-position. As the crank pin travels in the direction of rotation, the crank arm cuts the right valve circle, showing that the valve is moving to the right. At OA_H the valve displacement is equal to the steam lap, and as the valve is still moving to the right, the valve is at the point of opening, and A_H is the location of the crank pin at admission. As the crank continues its motion, the valve moves farther to the right, admitting steam into the cylinder until, at OM, the valve displacement is maximum. The valve, then, has moved to its extreme right-hand position. As the crank proceeds, the valve reverses its travel and moves to the left, and when the crank reaches the position OC_H, the flow of steam into the cylinder is cut off. With the crank located at OP', the valve is again at its mid-position, but it is traveling to the left. Then, at the position OR_H, the steam begins to discharge from the cylinder to the exhaust line, and at ON the valve has reached its extreme left-hand position, and starts its travel to the right. At the position OK_H, which is the point of compression, the valve has cut off the flow of steam from the cylinder. At OP, the cycle is completed.

In Fig. 171 d is shown the Zeuner diagram for the crank end of the same engine. As in the usual analysis, the crank-end diagram is superimposed upon the head-end diagram. The crank and eccentric circle, the valve circles, and the line of centers MN are common to both diagrams. In the crank-end diagram, the steam-lap arc is drawn intersecting the left valve circle, because the valve must move to the left to bring the steam edge of the valve over the edge of the crank-end port. Likewise, the arc, with the radius equal to the crank-end exhaust lap, is drawn, cutting the right valve circle.

The method of determining the events of the cycle, as percentages of the piston stroke or displacement, is shown for cut-off, for the head end, in Fig. 171 c. Considering the crank and eccentric circle as the crank circle, the circumference represents the path of travel of the crank pin, and the diameter, the travel of the crosshead or of the piston. With the crank pin at C_H, the piston position could be deter-

mined by drawing a perpendicular line from C_H to the horizontal XX', and this method may be followed, if extreme accuracy is not desired. However, for accurate work the angularity of the connecting rod must be considered, and the piston position is found by swinging an arc from C_H to C' on the line XX'. The radius used depends upon R/L, the ratio of the length of the radius of the crank arm to the length of the connecting rod. The radius of the crank arm is represented by the radius of the crank and eccentric circle, and if a value of R/L is assumed to be $\frac{1}{5}$, then the radius used to draw the arc C_HC' is equal to five times the length OC_H, and the center used lies along the diameter XX', extended past the head end of the circle. During the head-end stroke, the piston has moved from X to C_H', up to the time of cut-off. It should be noted that for the crank end the center used for the swinging arcs is on the horizontal diameter, extended past the head end of the diagram. However, the head-end events are measured from the head end of the diagram and the crank-end events from the other end. For example:

$$\left.\begin{aligned}\text{Head-end cut-off} &= 100 \times \left(\frac{XC_H'}{XX'}\right) = \text{percentage of stroke}\\ \text{Crank-end cut-off} &= 100 \times \left(\frac{X'C_C'}{X'X}\right) = \text{percentage of stroke}\end{aligned}\right\} \quad (117)$$

The other events are determined by the same method.

187. Reversing Valve Gears.—For certain classes of work, steam engines must be equipped with valve gears, by means of which the direction of rotation can be reversed easily and quickly. There are many different types of reversing valve gears. The principle of those commonly used is quite similar, in that the valve receives its motion from two separate mechanisms. The locomotive engine is the best known example of a reversible steam engine, and the Walschaert valve gear is used on most of the larger locomotives of this country, as well as being widely used in Europe where it was invented and developed.

The locomotive may be termed a duplex engine, since it consists of essentially two separate engines: one on each side of the frame, and each with a separate set of driving wheels. The eccentrics and driving cranks are set 90 deg. apart, thus making it impossible for the locomotive to stop with both cranks on the dead-center position. Each engine has a separate valve gear, and both are operated from a common reversing lever in the cab.

188. The Walschaert Valve Gear.—The Walschaert gear is placed on the side of the engine, outside of the frame. There are several advantages of this location: (1) the boiler may be made as large in

diameter as desired, which is not the case when the gear is placed under the boiler and between the driving wheels; (2) so placed, the gear is accessible for repairs and oiling; (3) it is away from most of the dust and grit drawn off the road bed.

Referring to Fig. 172, the eccentric crank *CE*, sometimes called return crank, is attached to the crank pin *C*. The reverse link receives a reciprocating motion from the eccentric pin *E*, which is the same as if an eccentric *OE* were used. The angle *COE* is 90 deg. The use of an eccentric crank is necessary because the eccentric rod is outside of the connecting rod. The reverse link is pivoted at the center and rocks about its pivot. Near one end of the radius rod is a block which fits in the slot of the reverse link and which can be raised or lowered in the slot by means of the lifting link. In Fig. 172 the block is shown

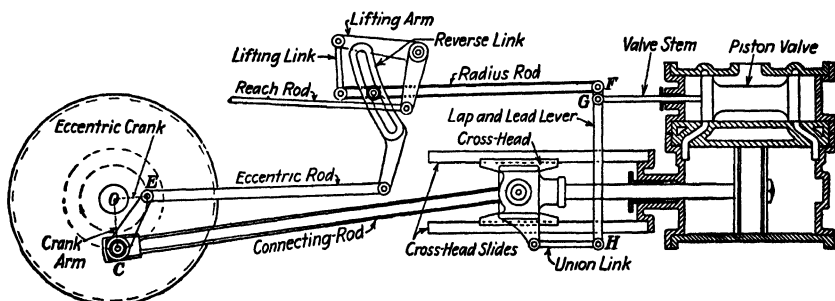



FIG. 172.—Diagrammatic layout of Walschaert valve gear, with a piston having inside admission.

in the middle of the link, and the radius rod is in mid-gear position. Points represent the reversing shaft by means of which the radius rods on both sides are operated. In the position shown, the valve would receive no motion from the radius rod, as it is at the center of the reverse link.

The lap-and-lead lever *FH* is a floating lever and receives a motion at the lower end from the travel of the crosshead. In this case, the upper end *F* acts as a pivot, and the point *G* receives a motion equal in length to twice the steam lap plus lead. As this motion is constant and also independent of the eccentric motion, it may be seen that when the piston is on either dead center, the eccentric is on its mid-position, and the valve is open a distance equal to the lead which is due to the effect of the lap-and-lead lever. Furthermore, this lead is constant for all positions of the radius rod.

When the radius rod is in the full-gear, forward position, the link block is at the bottom of the reverse-link slot, and the point *F* is given

maximum travel. This is the starting position under load and gives latest cut-off and maximum power per stroke in the cylinder. As the engine gains speed, the link block is moved nearer the middle of the link, bringing the cut-off earlier in the stroke, which results in using the steam more expansively.

The linkage is in the full-gear, backward position when the radius rod is lifted and the block is at the top of the slot in the reverse link.

189. Governing of Steam Engines.—The *function of a governor* is to maintain a constant speed despite any fluctuation that may occur in the load. There are certain steam-engine installations, such as steam-engine-driven air compressors and various water pumps, where the speed of the engine is permitted to vary considerably, and the governor regulates the speed of the engine to keep the discharge pressure of air or water constant. In cases where the steam engine is used to drive electrical generators or general machinery, it is important that the speed of rotation be kept constant, and it is in reference to this type of service that governors will be here discussed.

If there were no governing, in such cases, a decrease in the load on the engine would result in speeds which would cause excessive acceleration in the moving parts, and this could be continued indefinitely. And also, when the load increases, an engine would stop unless a governing device were provided to increase the steam supply.

Steam engines are built so that the energy supplied to the cylinder may be regulated in two ways, as follows:

1. By throttling, in which the steam pressure is varied, while the cut-off remains constant.

2. By changing the cut-off, in which the quantity of steam admitted to the cylinder is varied, while the pressure of the steam entering remains constant.

190. Speed Regulation.—Since a governor of either type requires a change in the speed before it will act to change the indicated power, there is necessarily a change in speed with fluctuating load. A well-designed governor will have a "close speed regulation"; that is, for the entire range of load from zero to full load there will be only a very small change in the speed, the speed being slightly decreased as the load increases.

The *coefficient of regulation* is the value obtained by dividing the r.p.m. of speed variation from the mean speed by the mean speed. If N_0 and N are the rotative speeds (r.p.m.) at zero load and full load, respectively, $N_0 - N$ equals the total speed variation, and the variation from the mean speed is half the total variation, or $\frac{N_0 - N}{2}$.

The mean speed regulation coefficient may be expressed by the following equation:

$$\text{S. R.} = \frac{N_0 - N}{2} \div \frac{N_0 + N}{2} = \frac{N_0 - N}{N_0 + N} \qquad (118)$$

191. Governor Types.—A typical *throttling governor* is illustrated in Fig. 173. The weights of this governor are mounted on a rotating sleeve which receives motion, through bevel gears and pulley, from the main engine shaft, and they are attached to short levers which act on a collar on the valve stem. The valve stem is placed inside the hollow sleeve extending down to the balanced governor valve which controls the flow of steam into the steam-engine valve chest. An adjustable spring opposes the downward force of the weights on the valve stem, and by changing the tension, the speed of the engine can be changed accordingly.

Fig. 173.—Throttling governor.

With a drop in load on the engine the speed increases. Owing to the increase in the centrifugal force of the weights, the spring resistance is overcome, and the valve stem is moved downward. This partially closes the balanced valve and reduces the flow of steam to the valve chest. For an increase in load, the valve is opened, increasing the steam supply and, likewise, the pressure.

The throttling governor is ordinarily used on simple, slide-valve engines which operate with the cut-off at about 50 per cent of the stroke.

Cut-off governors act directly on the valve gear, and they may be incorporated in the design of almost every type of steam engine. So-called *shaft governors* are of the cut-off type, various designs of which will be discussed in the following paragraphs.

Shaft governors are attached to an engine (usually the flywheel) so that their axis of rotation coincides with that of the main shaft. They depend, for functioning, on changes in centrifugal or inertia forces of attached weights, resulting from the rotation of the engine flywheel. In general, shaft governors may be divided into three classes: (1) centrifugal, (2) inertia, and (3) combined centrifugal and inertia.

The *Robb-Armstrong-Sweet centrifugal shaft governor* is illustrated in Fig. 174. In this design, a leaf spring is attached at one end to the rim of the flywheel. The other end carries a weight and is attached to

a counterweighted lever pivoted near the hub of the wheel. Between the pivot and counterweight, and mounted on the lever, is an eccentric pin which rotates with the flywheel and, through the eccentric rod, gives a reciprocating motion to the valve. When the engine is running, an increase in speed increases the centrifugal force of the weight and causes it to move outward, against the force of the spring. Moving outward, the weight turns the pivoted lever, thus changing the position of the eccentric pin with respect to the center of the shaft. This

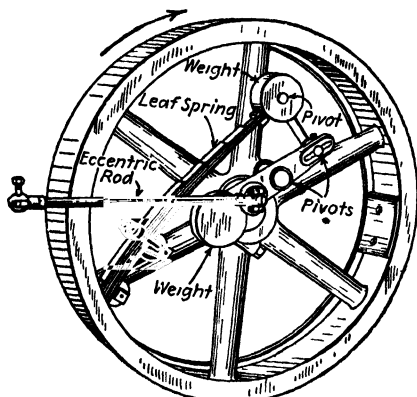



FIG. 174.—Robb-Armstrong-Sweet centrifugal governor.

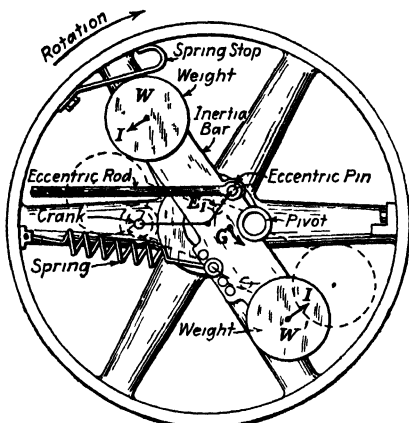



FIG. 175.—Rites inertia governor.

governor changes the valve travel and also the angular relation between the crank and eccentric.

The *Rites inertia shaft governor* is illustrated in Fig. 175. It has a weighted arm mounted on the flywheel and pivoted near the center of the shaft. The weight W' is heavier than W, which places the center of gravity of the arm at G. In the starting position, the coil spring holds the arm in the extreme position to obtain maximum cut-off. When the engine attains normal speed, the centrifugal force of the unbalanced weight arm overcomes the spring force and moves the arm around its pivot. The eccentric pin moves in the arc of a circle to a new position E_1, giving an earlier cut-off. The principal action of this governor, is, therefore, due to centrifugal force, but there is also an action due to inertia of the weight arm.

As the engine changes its speed, the weight arm tends to continue at the same speed of rotation because of its inertia. This causes the motion of the arm to lag behind that of the wheel. If the speed of the engine increases, the inertia forces of the weights, designated as I, will move the arm in the same direction around its pivot as the cen-

trifugal force acting at the center of gravity. This will immediately bring the cut-off earlier and tend to keep the speed regulated. If the flywheel decreases in speed, the inertia effect is in the opposite direction, changing the eccentric pin to a position to bring the cut-off later.

Figure 176 shows the Skinner unaflow engine and its shaft governor. This governor is a combination of the centrifugal and inertia types. The inertia arm is mounted on a roller bearing, and a change in the speed of the engine changes the position of the arm, relative

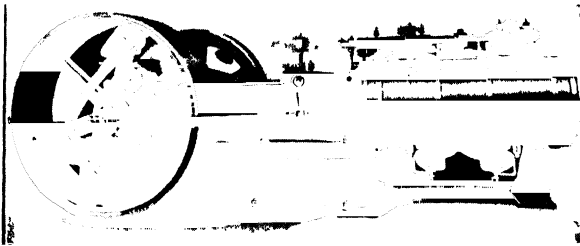



FIG. 176.—Skinner unaflow engine, showing governor.

to the flywheel. Through a ball-and-socket joint, eccentric rod, rocker arm and valve-operating mechanism, the valve action is changed, thus keeping the speed constant.

A good governor should have the following characteristics:

1. Closeness of regulation (small coefficient of regulation).
2. Should adjust quickly and positively.
3. Should be stable. (Unstable governors surge.)
4. Should be powerful enough to change position of valve.
5. Should have a safety device to stop the engine if the speed becomes excessive.

192. Engine Indicators.—The engine indicator is an instrument designed for obtaining a graphical record of the cylinder pressure as the piston of an engine moves through its cycle. Such a graph is of value in determining the events of the cycle, indicated or input horsepower, etc., and is commonly called an indicator diagram.

A typical indicator diagram for a simple, slide-valve steam engine is shown in Fig. 177. In this figure, the relation between the pressure and the piston displacement is clearly shown. It may also be seen that the indicator diagram is simply a pressure-displacement graph, its length representing the stroke of the engine and the height the pressure at all points of the stroke. Both of these dimensions are drawn to scale. The construction and operation of the indicator will be discussed in the following paragraphs.

Figure 178 illustrates a type of engine indicator that is widely used. It is designed for conditions of high temperature and pressure,

and consists, essentially, of two separate mechanisms. One has to do with the pressure, and is made up of the cylinder, piston, piston

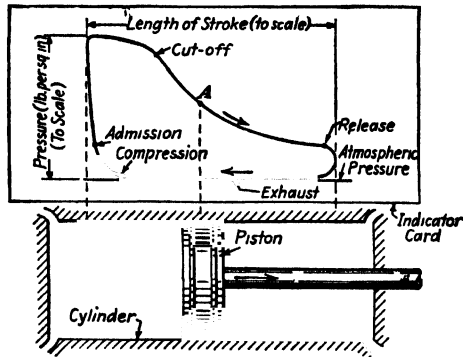



FIG. 177.—Illustrating the relation between the indicator diagram and the engine cylinder.

rod, spring, and the straight-line linkage which carries the marking pencil. The other consists of an oscillatory drum which holds the

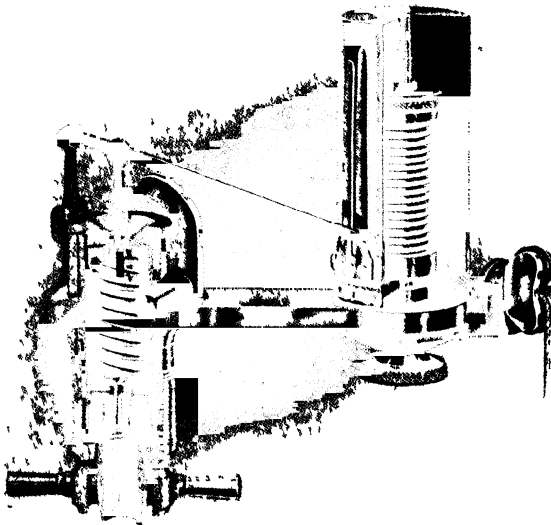



FIG. 178.—Crosby inside-spring steam-engine indicator.

card, drum spring and the cord. By means of the cord, the drum may be rotated around the vertical shaft. The drum spring causes the drum to reciprocate this motion.

The indicator piston has a definite area (usually 0.25, 0.50 or 1.00 sq. in.) and the spring above it is accurately made and calibrated for the piston area. The springs are of various sizes, depending on the engine-cylinder pressure for which it is used, and its *value* (10, 20, 40, 60, 80, 100, etc.) is such that when a pressure, in pounds per square inch, equal to the calibrated value is exerted in the indicator cylinder, the marking pencil will move, in a vertical straight line, just 1 in. Thus, a spring value of 40 indicates that when used with the proper-sized piston, the pencil will move up 1 in. for every 40 lb. per square inch range of pressure exerted in the cylinder.

The piston spring has one end screwed to the cylinder cap and the other secured to the piston, as shown. It may be removed by first removing the whole pencil mechanism. This is done by unscrewing the cylinder cap and lifting it upward. The spring proper is next unscrewed from the cap, which also releases the lower portion of the piston rod. The lower portion of the piston rod is then removed by unscrewing it from the piston. This releases the spring.

The paper cards generally used for indicator diagrams are treated on one side in a way so that this side will be marked by brass or other soft metal. This permits the use of a brass pencil or stylus which eliminates frequent sharpening.

In preparation for use, the indicator is mounted on a suitable cock at the end of a short pipe which is connected to the end of the engine cylinder. When the cock is open, engine-cylinder pressure is exerted in the cylinder of the indicator, and the pencil mechanism moves in sympathy with any variation. The drum cord is attached, by means of a suitable hook, to a reducing mechanism mounted on the engine, usually near the crosshead. With this mechanism, the stroke of the engine is reduced to the allowable circumferential travel of the drum. By the action of the drum spring and the motion of the engine, the drum is oscillated as desired.

In taking a diagram, a suitably sized card is placed on the drum. The cord is attached to the reducing mechanism, and the indicator cock is opened. The pencil is then thrust against the card for the interval of at least one complete cycle. After closing the cock and loosing the cord from the engine, the card may be removed.

There are many styles and types of indicators in use. The ordinary instrument is usually not considered accurate for speeds in excess of 300 r.p.m. There are indicators for use with speeds as high as 3,000 r.p.m. These, however, are generally used in internal-combustion engine practice. Some indicators are built with their piston springs enclosed, in the cylinder, and others, with the springs outside

and exposed to the atmosphere. The former type is illustrated in Fig. 178. Inside-spring indicators are generally affected by heat, and do not give the best obtainable accuracy under high temperatures.

There are also indicators available for taking so-called continuous diagrams. These are used in cases where an engine cannot be shop tested, and where the load is intermittent. Examples of such cases are rolling mill, hoisting and logging engines. Continuous diagrams

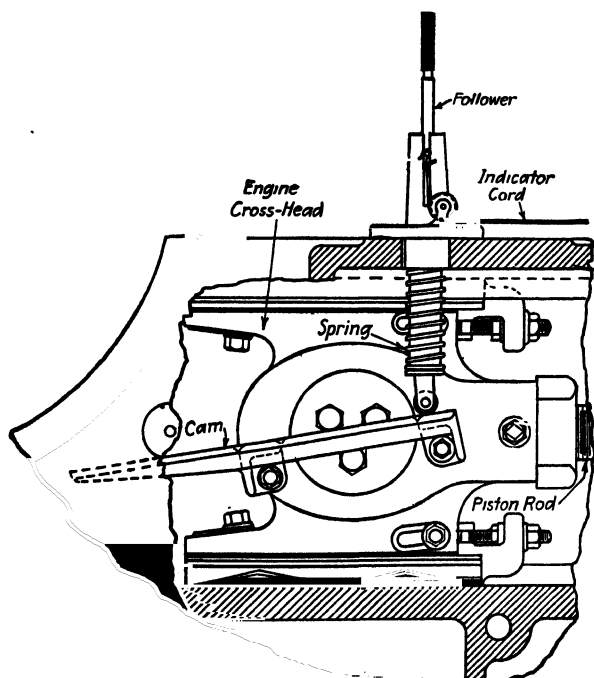



Fig. 179.—Steam-engine reducing motion.

are simply a series of diagrams, one behind the other, on a long strip of paper.

193. Stroke-reducing Mechanisms.—A type of stroke-reducing mechanism often used on steam engines is illustrated in Fig. 179. It consists of an inclined cam, fastened to the engine crosshead, and a vertical roller or follower rod. The follower is held in contact with the cam by a spring, as shown. As the crosshead moves back and forth, the follower oscillates vertically. The indicator cord is fastened to the follower, and it passes through the small pulley, as shown, to the indicator. The proportion of stroke reduction of this type of mechanism is determined by the inclination of the cam.

There are many different types of stroke-reducing mechanisms. The type best suited for a particular service depends on the engine speed and the accuracy of reduction desired.

194. The Planimeter.—It is generally necessary to obtain the area of the indicator diagram in order to determine the mean effective pressure for use in calculating the indicated horsepower. This may be done with a planimeter, one style of which is shown in Fig. 180. It consists, essentially, of two arms and a graduated roller wheel *D*. The diagram is traced in a clockwise direction with the point *P*, while the point *F* is fixed. When the arm *A* is of the proper length, the area, in square inches, may be read on the graduated wheel. The two arms should be at about 90 deg. when the planimeter is being used.

195. Mean Effective Pressure.—Practically all stationary steam engines are double acting; that is, the steam is admitted to both ends of the cylinder and acts on both sides of the piston. During the power

Fig. 180.—Planimeter.

stroke in the head end, steam is being exhausted from the crank end of the cylinder and *vice versa*. While the high-pressure steam is doing work on one side of the piston, the piston is doing work on the low-pressure steam on the other side. Work is equal to the product of the force times the distance. In the case of a single-acting cylinder, the force would be the average steam pressure, during the power stroke, of the piston times the area of the piston. But in the double-acting cylinder, this force is opposed by a negative force on the opposite side of the piston equal to the average steam pressure during the exhaust stroke.

To simplify calculation it may be assumed, with reasonable accuracy, that the negative work of the head end and crank end is equal. Then the net force acting on either end is determined by subtracting from the average pressure during the power stroke, the average pressure during the exhaust stroke for the same end. The work done at either end, then, is equal to the net average pressure times the net area of the piston.

The difference between the average pressures for the power stroke and exhaust stroke is called the mean effective pressure (m.e.p.).

It may be determined by first obtaining the area of the indicator diagram, in square inches (using a planimeter). This value is then divided by the length of the diagram, and the quotient multiplied by the scale or value of the indicator spring. Thus

$$P = \frac{a}{l} \times S \tag{119}$$

in which

P = mean effective pressure, lb. per square inch.
l = length of indicator diagram, in.
a = diagram area, sq. in.
S = spring value, lb. per square inch, per inch.

Another method of determining the mean effective pressure is by measuring and then averaging the ordinates on the diagram. Using this method, the diagram is divided by any convenient number of vertical lines equally spaced, as shown in Fig. 181. The vertical, dotted lines are then drawn, bisecting these areas, and the average length of these determined. This length is converted into pounds, giving the mean effective pressure. Thus, by this method,

FIG. 181.—Ordinate method of determining the mean effective pressure.

m.e.p. = average ordinate × scale of spring

The planimeter method is more accurate than the average ordinate method.

196. Indicated Horsepower.—The unit of one *engine horsepower*, as first used by James Watt, is equivalent to 33,000 ft.-lb. of work per minute. The indicated horsepower (i.hp.) is determined with the aid of the indicator diagram from the net work done by the steam inside the engine cylinder. The net force acting on the piston is the product of the mean effective pressure and the area of the piston. The distance through which the force acts in one minute is equal to the length of stroke times the number of strokes per minute, and the number of working strokes per minute, for an engine, bears a definite relation to the number of revolutions per minute (r.p.m.). For one end of a steam-engine cylinder, the expression for the indicated horsepower is as follows:

$$\text{i.hp.} = \frac{PLAN}{33{,}000} \tag{120}$$

in which

P = m.e.p., lb. per square inch.

L = length of piston stroke, ft.

A = net area of piston, sq. in.

N = strokes per minute (r.p.m., for one end).

There is usually a difference in the value of P for the head end and P for the crank end. The area of the piston rod must be subtracted from the piston area of the crank end; also, for the head end, if a tail rod projects through the head end of the cylinder.

Fairly accurate values of the *total indicated horsepower* for one cylinder may be determined as follows:

$$\text{total i.hp.} = \frac{2PLAN}{33{,}000} \tag{121}$$

In this case

P = average of m.e.p. (H.E. and C.E.).

In most cases, however, the total indicated horsepower is taken as the sum of the indicated horsepower of the two ends of the cylinder.

197. Brake Horsepower.—Owing to friction of the moving parts, the power delivered at the crank shaft is less than the indicated horse-

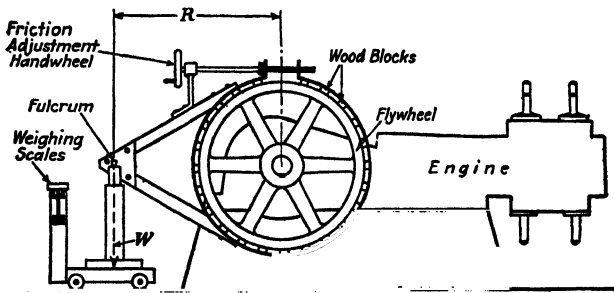



FIG. 182.—Prony brake.

power. The most common method of measuring the power delivered (brake horsepower, b.hp.) for medium-sized engines is by the use of a Prony brake. One type of Prony brake is illustrated in Fig. 182. It consists of metal strips holding a number of wooden blocks around the outside of the engine flywheel, thus forming a brake band. The brake band may be loosened or tightened by a hand screw. It is rigidly connected to a brake arm, as shown, at the end of which is a fulcrum which rests on a set of platform scales. Water is poured into the flywheel rim to absorb the heat of friction.

As a result of the brake friction, a downward force W is exerted and is measured by the scales. In Fig. 182, when the engine is run-

ning over, $W = w + w_o$, where w_o is the force due to the unbalanced weight of the brake arm, and w is the net force due to the friction of the brake.

$$\text{b.hp.} = \frac{2\pi RwN}{33{,}000} \tag{122}$$

in which

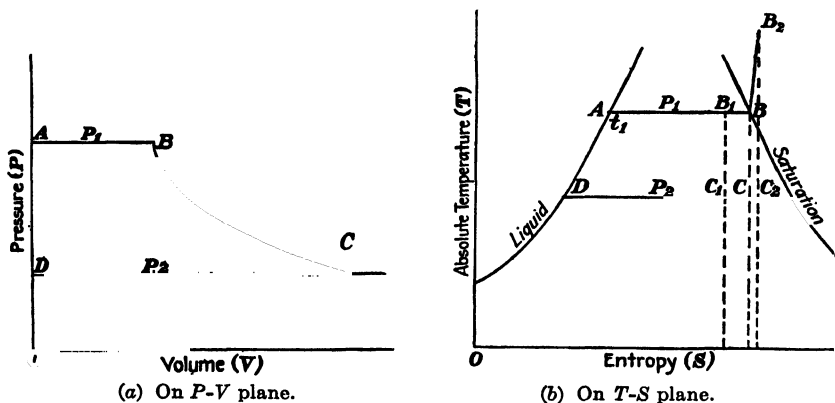
R = length of brake arm, ft.
w = net load on the scales, lb. $= W - w_o$.
N = r.p.m.

198. Friction Horsepower and Mechanical Efficiency.—The horsepower loss due to friction in the engine mechanism is called the friction horsepower. Friction horsepower may be determined as follows:

$$\text{f.hp.} = \text{total i.hp.} - \text{b.hp.}$$

The mechanical efficiency (e_m) is the ratio of the delivered horsepower to the total horsepower developed by the cylinder (horsepower input), and may be expressed as follows:

$$e_m = \frac{\text{b.hp.}}{\text{total i.hp.}} \tag{123}$$

199. Thermal Efficiency.—Rankine, a Scottish engineer, began his work on the theory of the transformation of heat into work in 1849.

(a) On P-V plane. (b) On T-S plane.
Fig. 183.—Theoretical Rankine cycle.

At about the same time, Clausius, a German physicist, independently covered the same ground and also developed the conception of entropy. Practice attempts to approach the ideal or *Rankine cycle* of operation with the actual steam-engine cycle.

In the ideal cycle (Fig. 183) a quantity of water at point A, under pressure p_1 and saturation temperature t_1, is admitted to the cylinder.

Heat is added at constant pressure and the water vaporizes, increasing the volume to B. Then expansion is permitted, and the steam expands at constant entropy to C which is at a pressure p_2. From C to D the steam is condensed to water at pressure p_2 and the corresponding saturation temperature. The water is then removed from the cylinder, placed under pressure p_1, heated to saturation temperature t_1 and returned to the cylinder under conditions corresponding to point A. The heat supplied to the cycle is the heat added to the water, from D to A, and to the steam, from A to B. The heat rejected by the cycle is the heat given up by the steam in condensing from C to D. The energy available for work may be clearly seen on the T-S diagram as the area $ABCD$ which is the difference between the heat content of the steam at B and at C. The thermal efficiency of the Rankine cycle is the ratio of the available energy to the heat supplied as follows:

$$e_R = \frac{h_1 - h_2}{h_1 - h_{f2}} \times 100 \qquad (124)$$

in which

e_R = thermal efficiency of Rankine cycle, per cent.
h_1 = enthalpy of 1 lb. of steam at start of constant-entropy expansion, B.t.u.
h_2 = enthalpy of 1 lb. of steam after constant-entropy expansion, B.t.u.
h_{f2} = enthalpy of liquid at exhaust pressure, B.t.u.

If the steam is wet saturated at the beginning of expansion, the cycle will be similar to AB_1C_1DA on the T-S diagram. In such a case, $h_1 = h_f + xh_{fg}$. If the steam is superheated, the cycle will be similar to AB_2C_2DA, and $h_1 = h_f + h_{fg} + c_p(t_s - t)$ where $(t_s - t)$ is the degrees of superheat. In either case, h_1 can be readily found in the steam tables or on the Mollier diagram. It should be noted that in the Rankine cycle heat is added, in the cylinder, along a constant pressure line rather than along an isothermal line.

With a high pressure difference $(p_1 - p_2)$, the cycle efficiency, to be accurate, must include the *pump work*. Thus, Eq. (124) becomes

$$e_R = \frac{h_1 - h_2 - A144\,(p_1 - p_2)v_{f2}}{h_1 - h_{f2} - A144\,(p_1 - p_2)v_{f2}} \times 100 \qquad (124a)$$

200. Thermal Efficiency with Incomplete Expansion.—In the Rankine cycle with incomplete expansion, the expansion is not carried to the final exhaust pressure but is stopped at some intermediate pressure p_3. From p_3 there is a constant volume change to the exhaust

pressure p_2. While this cycle has a lower efficiency than the complete Rankine cycle, it is more like the cycle of the actual steam engine. Complete expansion down to the exhaust pressure would necessitate extremely large cylinder volumes and would be impractical. The small gain in efficiency would be more than offset by the increased cost of the large cylinders.

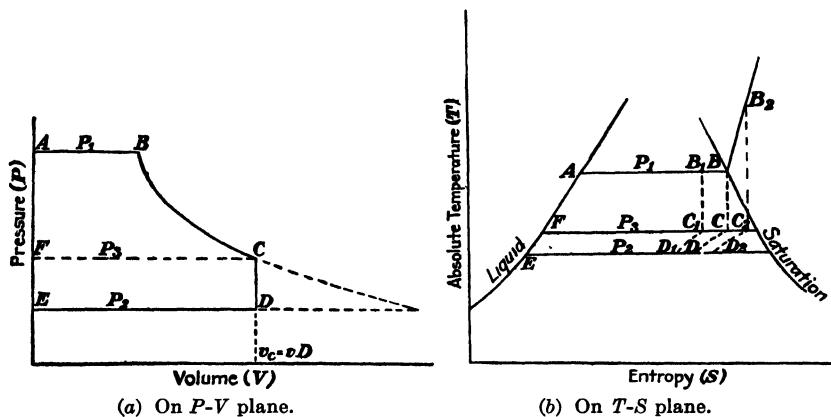
From the *P-V* diagram (Fig. 184) it may be seen that the work of this cycle is made up of two areas, *ABCFA* and *FCDEF*. Area *ABCFA* represents the work of a Rankine cycle between the pressures p_1 and p_3 and may be expressed in B.t.u. per pound of steam, as $(h_1 - h_3)$. Area *FCDEF* is the difference between the work done at

(*a*) On *P-V* plane. (*b*) On *T-S* plane.

FIG. 184.—Theoretical Rankine cycle with incomplete expansion.

constant pressure, from *F* to *C*, and the constant-pressure work from *D* to *E*. In foot-pounds, this is equal to $144(p_3v_C - p_2v_D)$. As $v_C = v_D$, this work is equal to $144(p_3 - p_2)v_3$.

The thermal efficiency (omitting pump work) is as follows:

$$e = \frac{h_1 - h_3 + A144(p_3 - p_2)v_3}{h_1 - h_{f2}} \times 100 \qquad (125)$$

in which

e = thermal efficiency, per cent.
h_1 = enthalpy, B.t.u. per pound steam at p_1.
h_3 = enthalpy, B.t.u. per pound steam at p_3 (after constant-entropy expansion).
p_2 and p_3 = pressures, lb. per square inch absolute.
v_3 = volume per pound of steam at p_3, cu. ft.

Pump work is included as shown by Eq. (124*a*).

201. Actual Thermal Efficiency.—If there were no losses in the cylinder, all of the B.t.u. theoretically made available in the heat

engine cylinder would be transformed into work and would be evident in the indicated horsepower. Because of cylinder losses, the ratio of the indicated work, in B.t.u., to the heat supplied gives the *actual thermal efficiency*. Thus,

$$\text{actual } e_t = \frac{\text{indicated work, B.t.u.}}{\text{heat supplied to cylinder}} \tag{126}$$

Since 2,545 B.t.u. are equivalent to 1 hp.-hr.,

$$e_t = \frac{\text{i.hp.} \times 2{,}545}{\text{lb. steam per hr. } (h_1 - h_{f2})} \tag{127}$$

or

$$e_t = \frac{2{,}545}{w(h_1 - h_{f2})} \tag{128}$$

in which

w = water rate, pound steam per indicated horsepower-hour.

Other symbols are as before.

In many cases it is useful to determine the *thermal efficiency based on the brake horsepower*. On this basis,

$$e_t = \frac{\text{b.hp.} \times 2{,}545}{\text{lb. steam per hr. } (h_1 - h_{f2})} \tag{129}$$

If the engine is direct connected to a generator, the output of which is measured in kilowatts (kw.), the thermal efficiency of the unit, based upon the power delivered, is equal to

$$e_t = \frac{\text{kw.} \times 3{,}413}{\text{lb. steam per hr. } (h_1 - h_{f2})} \tag{130}$$

The *heat consumption per hour* and *heat supplied per hour* have the same meaning and are used interchangeably. The heat consumption per horsepower per hour $= w(h_1 - h_{f2})$ if w = water rate, pound steam per horsepower-hour. This may be based on indicated or brake horsepower, as desired.

202. Efficiency Ratio.—The ratio of the work actually done to the work done in the theoretical cycle is a measure of the perfection of design and construction of the actual engine. This ratio, termed the *efficiency ratio*, or *engine efficiency*, may be determined by dividing the thermal efficiency of the actual engine by the theoretical cycle efficiency. Usually the Rankine cycle efficiency is used for obtaining the efficiency ratio. For practical reasons steam engines are not designed to operate on the Carnot cycle. It is often desirable to

compare steam engines with steam turbines in which the expansion is completed to the exhaust pressure. Because they operate thus, the Rankine cycle with incomplete expansion does not apply to the turbine.

$$\text{efficiency ratio} = \frac{e_t \text{ (actual)}}{e_R \text{ (Rankine cycle)}} = \frac{2{,}545 \times \text{i.hp.}}{\text{lb. steam per hr. } (h_1 - h_2)} \qquad (131)$$

This efficiency ratio is sometimes termed the *Rankine cycle ratio.*

Example 10-1.—A 12- by 18-in., simple steam engine, with a piston rod 1.8 in. in diameter, gave the following test data; speed 265 r.p.m.; area indicator diagrams H.E., 1.85 sq. in., C.E. 1.92 sq. in.; length of diagram 3.2 in.; indicator spring scale 80 lb.; the length of Prony brake arm is 65 in.; the net brake load is 359 lb.; steam pressure at the throttle, 155 lb. per square inch gage; quality 98.7 per cent; exhaust pressure 4.5 lb. per square inch gage; pressure at release 47 lb. per square inch gage; barometer 29.5 in. of mercury; water rate, 29.2 lb. of steam per indicated horsepower-hour. Calculate (*a*) the mechanical efficiency, (*b*) the thermal efficiency based on the indicated horsepower, (*c*) the theoretical efficiency on the Rankine cycle and (*d*) on the incomplete Rankine cycle.

Solution.—*a. The Mechanical Efficiency.*

$$\text{H.E. m.e.p.} = \frac{1.85}{3.2} \times 80 = 46.2 \text{ lb. per square inch}$$

$$\text{C.E. m.e.p.} = \frac{1.92}{3.2} \times 80 = 48 \text{ lb. per square inch}$$

$$\text{H.E. i.hp.} = \frac{46.2 \times 1.5 \times 113.1 \times 265}{33{,}000} = 63.0$$

$$\text{C.E. i.hp.} = \frac{48 \times 1.5 \times 110.6 \times 265}{33{,}000} = 63.9$$

$$\text{total indicated horsepower} = 126.9$$

$$\text{b.hp.} = \frac{2\pi 5.417 \times 359 \times 265}{33{,}000} = 98.4$$

$$\text{mechanical efficiency} = \frac{98.4}{126.9} \times 100 = 77.5 \text{ per cent}$$

b. Thermal Efficiency.

$$p_1 = 155 + 14.5 = 169.5 \text{ lb. per square inch absolute.}$$

$$p_2 = 4.5 + 14.5 = 19 \text{ lb. per square inch absolute.}$$

$$e_t = \frac{2{,}545}{29.2(340.77 + 0.987 \times 854.6 - 193.34)} = 0.088 \text{ or } 8.8 \text{ per cent}$$

c. Efficiency of Rankine Cycle (omitting pump work).

$$s_1 = 0.5265 + 0.987 \times 1.0324 = 1.5455$$

$$s_2 = 1.5455 = 0.3316 + x_2 1.4042 \qquad x_2 = 0.865$$

$$h_2 = 193.34 + 0.865 \times 961.7 = 1{,}024.70 \text{ B.t.u.}$$

$$e_R = \frac{1{,}184.27 - 1{,}024.70}{990.93} = 0.161 \text{ or } 16.1 \text{ per cent}$$

d. Efficiency of Incomplete Rankine Cycle (omitting pump work).

$$p_3 = 47 + 14.5 = 61.5 \text{ lb. per square inch absolute}$$
$$s_3 = 1.5455 = 0.4293 + x_3 1.212 \qquad x_3 = 0.921$$
$$h_3 = 263.64 + 0.921 \times 913.8 = 1{,}105.21 \text{ B.t.u.}$$
$$v_3 = 0.921 \times 7.01 = 6.46 \text{ cu. ft. per pound}$$
$$e_{IR} = \frac{(1{,}184.27 - 1{,}105.21) + [1/778 \times 144(61.5 - 19) \times 6.46]}{990.93}$$
$$= 0.1311 \text{ or } 13.11 \text{ per cent}$$

203. Theoretical Indicator Diagram.—When the actual indicator diagram is not available, but the conditions of pressure and volume for the cylinder are known, the theoretical diagram can be drawn, and the theoretical mean effective pressure can then be determined. The theoretical diagram represents the work diagram of an ideal engine operating under actual conditions.

In the construction of this diagram, the expansion line is drawn as an equilateral hyperbola ($pv = C$), this line being easily constructed and being very close to the true expansion curve on the *P-V* diagram. The ideal engine is also assumed to have zero clearance volume and 100 per cent release.

For the construction of the theoretical indicator diagram (Fig. 185), the following data are given:

p_1 = initial pressure, lb. per square inch absolute.

v_1 = volume of cylinder at cut-off (start of expansion), cu. ft.

v_2 = volume of piston displacement, cu. ft.

p_2 = exhaust pressure, lb. per square inch absolute.

FIG. 185.—Showing method of constructing the theoretical indicator diagram.

The ordinate lines, zero pressure and zero volume, are drawn. Then, to a chosen pressure scale, the line *AG* is drawn, to represent pressure p_1, and *ED* is drawn to represent p_2. Line *GF* is next drawn, using a convenient volume scale to represent v_2, and *BH* to represent v_1. The method used in constructing the equilateral hyperbola is given in the following paragraph.

Draw line *OG* to intersect the vertical line through *B* at *M*. Then, draw *MC*, a horizontal line through *M*; this represents the pressure at the end of expansion, p_3. Other points on the curve *BC* may be found by taking points, such as *P* on *BG*, and drawing the diagonal

OP to the origin, intersecting *BM* at *R*. Draw the vertical line through *P* and a horizontal line through *R*; their intersection *x* is a point on the curve.

The work of this ideal diagram is represented by the area *ABCDE*. The definite relations are given in the following:

$$\begin{aligned} \text{Area } ABCDE &= OABH + HBCF - OEDF \\ OABH &= p_1v_1 \\ HBCF &= p_1v_1 \log_e \frac{v_2}{v_1} \\ OEDF &= p_2v_2 \\ \text{Work in foot-pounds} &= p_1v_1 + p_1v_1 \log_e \frac{v_2}{v_1} - p_2v_2 \end{aligned}$$

If p_t = mean effective pressure (theoretical), and $r = v_2 \div v_1$ (ratio of expansion), then

$$\begin{aligned} p_t &= \frac{p_1v_1 + p_1v_1 \log_e r - p_2v_2}{v_2} \\ &= \frac{p_1v_1 + p_1v_1 \log_e r}{v_2} - p_2 \end{aligned}$$

Dividing the numerator and the denominator of this fraction by v_1,

$$\begin{aligned} p_t &= \frac{p_1(1 + \log_e r)}{\frac{v_2}{v_1}} - p_2 \\ &= \frac{p_1(1 + \log_e r)}{r} - p_2 \qquad (132) \end{aligned}$$

As mentioned before, the ideal engine is assumed to have zero clearance volume, and 100 per cent release. For this reason, on the theoretical indicator diagram, the ratio of expansion, *r*, is taken as the ratio of the volume of piston displacement to the part of the piston-displacement volume filled with steam at the cut-off point. The following expression is used to determine the ratio of expansion in calculating theoretical mean effective pressure.

$$r = \frac{100}{\text{per cent cut-off}}$$

If the ideal engine is assumed to have clearance, as accepted by many authorities, then the value of *r* is changed. Both v_1 and v_2 are increased by the clearance volume. The ratio of the actual mean effective pressure to the theoretical mean effective pressure is termed

the *diagram factor*. The following tabulation gives approximate values for the diagram factors for various types of steam engines.

Type	Simple	Compound
Single-valve, high-speed engine	0.80	0.70
Four-valve, high-speed engine	0.85	0.75
Corliss engine	0.90	0.80

204. Probable Mean Effective Pressure.—The probable mean effective pressure is an estimated value that may be obtained by multiplying the theoretical mean effective pressure by the diagram factor. Thus,

$$p_m = f \times p_t$$

in which

p_m = the probable mean effective pressure.
p_t = the theoretical mean effective pressure.
f = diagram factor.

205. Probable Indicated Horsepower.—The probable indicated horsepower may be determined by the usual formula, as follows:

$$\text{i.hp.} = 2 \times \frac{p_m LAN}{33{,}000} \quad \text{(for double-acting engine)} \qquad (133)$$

in which

p_m = the probable mean effective pressure, from Art. 204, and other symbols are as before.

206. Water Rate.—The actual water rate of an engine is determined by dividing the number of pounds of steam supplied per hour by the indicated horsepower, or brake horsepower, of the engine. In order to determine the weight of steam supplied per hour, and the horsepower, accurately, it is necessary to run an actual test. The steam used is condensed and this water weighed. All condensate from jackets, glands and drips should be included. If impossible to condense the steam, the feedwater to a boiler whose only outlet is to the engine under test is weighed.

Example 10-2.—From a head-end steam-engine indicator diagram taken with a 100-lb. spring, the following measurements were taken: height of maximum steam pressure above the atmosphere line 1.65 in.; height of exhaust pressure above atmosphere line 0.2 in.; length of diagram 3.58 in.; length to point of cut-off 1.62 in. The barometer is 29 in. of mercury; engine is double acting, having an 8- by 14-in. cylinder; speed, 185 r.p.m. Calculate the theoretical mean effective pressure, and, assuming that the mean effective pressure of both ends of the cylinder is the same, calculate the probable indicated horsepower using a diagram factor of 0.78.

Solution.—*Theoretical Mean Effective Pressure.*

$$p_1 = 1.65 \times 100 + 14.25 = 179.25 \text{ lb. per square inch absolute}$$
$$p_2 = 0.2 \times 100 + 14.25 = 34.25 \text{ lb. per square inch absolute}$$
$$r = \frac{3.58}{1.62} = 2.21, \qquad \log_e 2.21 = 0.793$$
$$p_t = \frac{179.25(1 + 0.793)}{2.21} - 34.25 = 111.5 \text{ lb. per square inch}$$
$$\text{Probable i.hp.} = \frac{2 \times 0.78 \times 111.5 \times 1.168 \times 50\,265 \times 185}{33{,}000} = 57.2$$

207. Steam-engine Losses.—The main sources of heat loss in a steam engine are those due to

1. Initial cylinder condensation.
2. Reevaporation
3. Wire drawing
4. Incomplete expansion.
5. Exhaust steam.
6. Radiation.
7. Leakage.

The loss due to *initial cylinder condensation* has been carefully studied since the time James Watt devised the separate condenser. The unaflow engine, to be taken up in the following chapter, is a result of the effort to overcome this loss.

At the time the inlet valve of a steam-engine cylinder opens, the cylinder is relatively cool, owing to the effect of the exhaust steam. Consequently, there is a flow of heat from the steam to the piston and cylinder walls. This loss is reduced in compound engines because of the reduction of the temperature difference of the entering and exhaust steam in each cylinder. High-speed engines tend to have less initial condensation loss due to the shorter time allowed for cooling. Steam jackets reduce the loss, and with the unaflow cylinder it is reduced to a minimum by eliminating the reverse-flow steam.

As the steam expands, its temperature drops below that of the cylinder walls, and it receives heat from the walls. As the exhaust valve opens, the pressure drops, and most of the water in the steam is *reevaporated.* Heat thus absorbed by the steam is of no use to the engine as it immediately passes out the exhaust. Jackets of live steam around the cylinder sides would merely increase this loss.

The loss due to *wire drawing* or *throttling* occurs when the valve is closing and when the steam flows through narrow passages. The throttling effect reduces the steam pressure and decreases the available energy of the steam. This loss is greatest in an engine having a slow-moving slide valve. The use of poppet or Corliss valves, or slide valves with more than one port, will reduce the wire-drawing loss.

The heat loss due to *incomplete expansion is comparatively small.* If steam were completely expanded to the exhaust pressure, a small increase in work would be obtained. However, the length of the

TABLE 10-1.—STEAM CONSUMPTION OF VARIOUS TYPES OF ENGINES

Type	Lb. of steam per horsepower per hour	Steam press. lb. per square inch, gage
High speed, simple, non-condensing	32	80 to 100
High speed, compound, non-condensing	24 to 26	150 to 210
High speed, compound, condensing	19 to 21	150 to 210
Corliss, simple, non-condensing	26	80 to 100
Corliss, simple, condensing	21	80 to 100
Corliss, compound, non-condensing	20 to 22	150 to 210
Corliss, compound, condensing	14 to 15	150 to 225
Triple expansion, condensing	13	150

cylinder would have to be greatly increased to provide for the increased volume of the steam. This would mean higher first cost and a bulky and heavy engine. Consequently, this loss is tolerated, since it is more than balanced by the improvement in the design of the engine.

TABLE 10-2.—WATER RATES OF UNAFLOW ENGINES AT VARIOUS LOADS

Rated capacity	R.p.m.	Initial press. lb. Abs.	Back pressure, lb. or vacuum, in. Hg.	Super heat, °F.	Lb. steam per i.hp., per cent load				
					25	50	75	100	125
150 kva	257	110	2 lb.	0	22.5	21.9	21.5	21.7	22.1
150 kva	257	110	2 lb.	0	21.6	20.8	20.6	21.1	
150 kva	225	110	2 lb.	0	24.8	22.4	21.8	21.9	
150 kva	225	110	2 lb.	0	23.9	23.2	22.8	22.6	
200 kw	150	200	1.25 lb.	0	....	....	....	18.5	19.1
250 kw	...	140	1 lb.	0	18.6	18.4	18.3	18.8	
250 kw	...	140	26 in.	0	13.6	13.5	13.7	13.8	
400 kw	200	200	1.25 lb.	0	....	....	....	19.7	20.3
525 kw	...	150	2 lb.	0	....	....	....	19	
525 kw	...	150	26 in.	0	....	....	....	14.1	
525 kw	...	150	26 in.	150	....	....	....	11.2	

The loss in the *exhaust steam* is unavoidable, but it can be reduced considerably by condensing operation. It is always the largest loss of any, but it can be offset, somewhat, by making economical use of the exhaust steam.

Leakage includes leakage of steam through valves, packings and around the piston head. The reduction of this loss is largely a matter of maintenance, replacing worn packings and rings, and adjusting valves.

In addition to the above-mentioned heat losses, there is the mechanical or *friction loss* to be considered. This, however, is comparatively small in well-constructed engines. The friction of an engine can be reduced to a minimum by using suitable oils for the lubrication of the piston and cylinder and other friction elements.

208. Steam-engine Economy.—Steam economy of engines is usually based on the weight of steam used per indicated horsepower per hour. A more accurate knowledge of the economy at which an engine operates may be obtained from the rate of heat consumption, expressed as the number of B.t.u. per indicated horsepower, or brake horsepower, per minute.

Economy, during operation, varies greatly with the relative load on the engine, and, also, on the steam conditions. Tables 10-1 and 10-2 give values for the economy of typical engines, at various loads of operation.

Problems

(All engines double acting)

1. An 8- by 14-in. steam engine, with connecting rod 42 in. long, has a valve travel of 3 in., and is designed to rotate, or run, *over*. Other data include; H.E. steam lap ¾ in.; H.E. lead 1⁄16 in.; H.E. compression 20 per cent; C.E exhaust lap ⅜ in. Use a 6-in. crank and eccentric circle and draw a Zeuner diagram and tabulate all results, assuming the sums of the steam and exhaust laps to be equal.

2. The following data were taken from an engine (running over): valve travel 3 in.; steam lap H.E. ⅞ in., C.E. ¾ in.; exhaust lap H.E. 3⁄16 in., C.E. 5⁄16 in.; angle of advance 35 deg.; R/L = ⅕. Draw Zeuner diagram and tabulate all results. Use a crank and eccentric circle 6 in. in diameter.

3. Find the indicated horsepower of a 10- by 14-in., 200 r.p.m. engine, having a piston rod diameter of 2½ in. The indicator diagram, taken with a 60-lb. spring, has an area of 2.85 sq. in., H.E.; 2.89 sq. in., C.E.; and a length of 3 in.

4. A 14- by 22-in. engine having a piston rod 2⅞ in. in diameter, and running at 175 r.p.m., delivers 100 b.hp. The average mean effective pressure is 37.5 lb. Find (*a*) the mechanical efficiency, and (*b*) the friction horsepower.

5. When the engine in Problem 4 operates condensing, the mean effective pressure is increased 48 lb. With the other conditions as before and with the same friction horsepower, determine (*a*) the indicated horsepower, (*b*) the mechanical efficiency.

6. An engine having a mechanical efficiency of 87 per cent, requires 3,200 lb. of steam per hour, while delivering 125 b.hp. Initial steam pressure is 175 lb. per square inch absolute; quality 98.5 per cent; exhaust pressure 2 lb. per square

inch absolute. Find (*a*) the thermal efficiency, based on the indicated horsepower, and (*b*) the heat supplied per brake horsepower per minute.

7. Solve Problem 6 when the engine operates under the conditions as given, except that the steam consumption is 2,800 lb. per hour; steam pressure 200 lb. per square inch absolute, at 150° of superheat.

8. An engine, operating on the incomplete Rankine cycle, uses steam at a pressure of 155 lb. per square inch absolute and 150° of superheat; release pressure is 35 lb. per square inch absolute; exhaust to a vacuum of 27 in. of mercury; barometer 30 in. of mercury. Determine (*a*) the theoretical work of the cycle, and (*b*) the water rate of the ideal engine.

9. Solve Problem 8 if the steam is 98 per cent dry, and other conditions are unchanged.

10. An engine operating on the Rankine cycle uses steam at a pressure of 178 lb. per square inch absolute; temperature 530°F.; exhaust pressure 15 lb. per square inch absolute. Find (*a*) the theoretical water rate and (*b*) the cycle efficiency; (*c*) determine, also, the actual steam used per brake horsepower per hour, and the thermal efficiency based on the brake horsepower if the engine uses 27,500 lb. of steam during 10 hours, while delivering 110 b.hp.

11. For a 16- by 24-in. engine, with a piston rod 3.25 in. in diameter, determine the H.E. and C.E. piston displacement in cubic feet.

12. If it takes 11.2 lb. of water at 75°F. to fill the H.E. clearance space, and 11.75 lb. to fill the C.E. clearance space of the engine of Problem 11, determine the percentage of clearance volume for each end.

13. Find the weight of dry steam in the cylinder at release for an engine 18- by 28 in.; release 85 per cent; clearance 9.2 per cent; and pressure at release 54 lb. per square inch absolute.

14. A 14- by 30-in. steam engine has a 2.5-in. piston rod and operates at 100 r.p.m.; has a net brake load of 1,050 lb.; brake arm 5 ft. From the indicator cards, the following data were taken: Length 3.24 in.; H.E. area 2.42 sq. in.; C.E. area 2.58 sq. in.; scale of spring 80 lb.; release 88 per cent; compression 21 per cent; clearance 5 per cent; H.E. release pressure 63 lb. per square inch absolute; exhaust pressure 21 lb. per square inch absolute. Calculate (*a*) the brake horsepower, (*b*) indicated horsepower, and (*c*) mechanical efficiency.

15. From the data of Problem 14, compute the weight of steam per indicated horsepower per hour, assuming the quality to be 100 per cent at release and compression.

16. A theoretical indicator diagram gives the following data: initial pressure 128 lb. per square inch gage; exhaust pressure 3.5 lb. per square inch gage; barometer 28.8 in. of mercury; cut-off is at 20 per cent of the stroke. Determine the theoretical mean effective pressure.

17. With a valve travel of 3 in.; lead $\frac{3}{16}$ in.; compression 20 per cent; cut-off 55 per cent, draw a Zeuner diagram for the head end, and tabulate all results.

CHAPTER XI

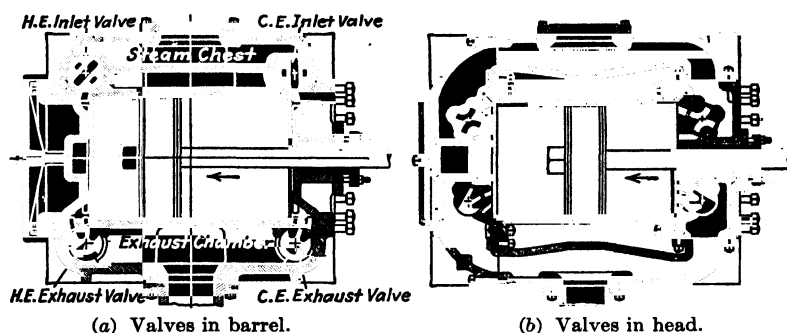
SPECIAL DESIGNS OF STEAM ENGINES

209. Introductory.—The steam engine, from its early design, has been developed along various lines which have tended toward the elimination of its original outstanding defects. This development has given rise to a variety of special designs, including the Corliss, unaflow, and compound engines, any of which have many points of superiority over the simple design of steam engine. These special types of engines will be taken up in the following articles.

210. Corliss Engine.—In 1846 George H. Corliss invented the four cylinder valves and the releasing valve gear used on the engine which is universally given his name. Many engine builders now make Corliss engines. The advantage of this type, as compared to the usual slide-valve engine, lies in its improved steam economy, which is due (1) to quick closing of the steam valve at cut-off, and (2) to small clearance volume. The sharp cut-off reduces the throttling effect of the steam entering the cylinder, and this results in increasing the available energy obtained from each pound of steam. The clearance volume is reduced by having the valves placed close to the cylinder and, thus, eliminating long steam ports.

The Corliss engine is built with either the releasing or the non-releasing valve gear. The non-releasing toggle and cam-operating valve gears are necessary for high-speed engines of the Corliss type. Two eccentrics are required, one for the two admission valves, and one for the two exhaust valves. Governing is accomplished by varying the eccentricity and the angle between the crank and eccentric. The cams or toggles are designed to permit quick opening and closing of the admission valves, allowing the valves to remain at rest during the period of greatest unbalance of pressure.

211. Corliss Releasing Valve Gear.—In the releasing valve gear, there is one steam valve and one exhaust valve at each end of the cylinder (Fig. 186). These valves are usually double ported and are placed either in the cylinder head or in the cylinder barrel, as shown in Fig. 186. There is one eccentric on the crank shaft, as shown in Fig. 187, which, through an eccentric rod, rocker arm, and reach rod, imparts an oscillatory motion to a wrist plate mounted on the side of the cylinder.

Four rods are attached to the wrist plate. The two lower rods are attached to arms of the exhaust valve stems. The exhaust valves,

(a) Valves in barrel. (b) Valves in head.
FIG. 186.—Sectional views of cylinders of Erie Ball Corliss engines.

therefore, are opened and closed, positively, by the eccentric, through the wrist plate. The events controlled by the exhaust valve, release

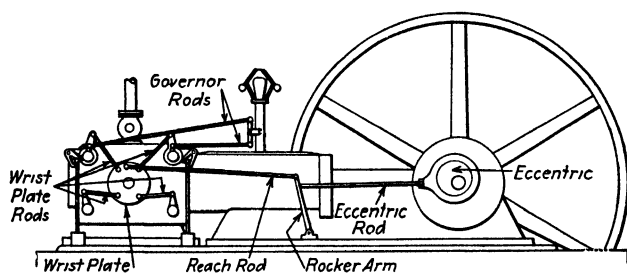



FIG. 187.—Diagrammatic sketch showing Corliss valve gear.

and compression are, therefore, constant and not affected by the governor action.

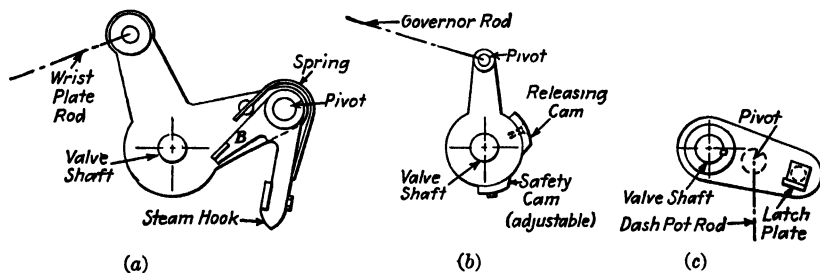



FIG. 188.—Details of Corliss releasing gear.

The steam valve gear is somewhat more complicated and is shown in detail in Fig. 188. Each valve stem projects through the side of the cylinder, and contains three independent cranks. The inside and middle cranks are loose and free to rotate, and the outside crank is

keyed to the stem. The wrist-plate rod is attached to one of the two arms on the inside rocker (Fig. 188 *a*), and on the other arm is mounted the steam hook. The arm *B* of the steam hook is forced against the middle crank by a spring. Figure 188 *b* shows the middle crank which contains the releasing cam, the safety cam and also an arm to which is attached the governor rod. The outside crank (Fig. 188 *c*) is keyed to the valve stem and has an arm called the steam arm. It has a latch plate, which engages with the steam hook, and also a connection to the dashpot. The complete assembly is shown in Fig. 189.

In the operation of the releasing valve gear, the top of the wrist plate moves toward the steam valve, the inside rocker moves clockwise (Fig. 189), and the steam hook is moved downward until it engages the latch plate on the steam arm. Then, as the wrist plate reverses its motion, the hook raises the steam arm, thus open-

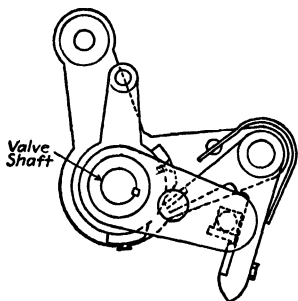



FIG. 189.—Assembly of Corliss releasing gear.

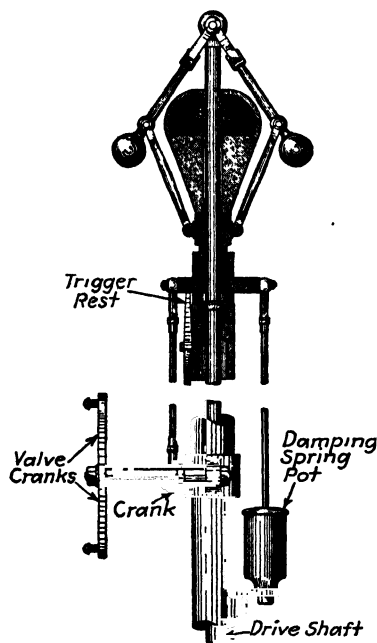



FIG. 190.—Corliss pendulum governor.

ing the admission valve. At the same time the piston in the dashpot is raised, creating a vacuum in the dashpot cylinder. When the arm of the steam hook strikes the releasing cam, the hook is disengaged from the latch plate, and the dashpot rod quickly pulls the steam arm down, closing the valve.

212. The Pendulum Type of Governor.—This type of governor is often used on Corliss engines, and is illustrated in Fig. 190. It is driven from the crank shaft by a belt. When the engine load is decreased, the weights move outward. This motion is transmitted, through a rocker arm, to the governor crank levers on the steam valves, which move the releasing cams to a position where the time of release of the

steam arm is earlier. Thus, with decreasing load, the cut-off is effected earlier in the stroke. The only event of the cycle that is changed by the governor is the cut-off.

If the governor belt breaks, the weights will drop down to a position that would correspond to the maximum-load position. The releasing cams on the steam valves would be moved back to a position where the steam hook would not be disengaged, and it is probable that the engine would "run away" and cause damage. To care for this, there is placed on the bottom of each governor collar a safety cam which may be brought into the proper position to prevent the steam hook from picking up the latch plate on the steam arm, when the governor weights are down. As a result of this, the steam valves will not open, and the engine, not receiving steam, will stop.

When starting the engine, the governor weights are lifted until the governor cross-bar rests on a trigger (safety stop) which is moved into position manually, or automatically by steam pressure.

213. Unaflow Engine.—The steam turbine was first introduced in America in 1900, and, due to its low cost, small size and better steam economy, as compared with reciprocating steam engines of that date, it appeared that the steam engine would be entirely superseded, except, perhaps, in the very small sizes. The turbine eliminated almost entirely the heat losses due to initial condensation and reevaporation, inherent in the steam engine, and proved better able to handle large volumes of steam for condensing operation.

The engine builders had made many improvements: the four-valve engine, the method of cylinder jacketing with live steam, and the multiple-cylinder expansion, but the common cylinder losses still remained. Prospects for the steam-engine industry did not appear promising.

About 1900, the *unaflow* engine was being developed and placed on a commercial basis in Europe, as a result of the work of Prof. J. Stumpf, of Germany, and N. W. Todd, of England. Soon after, this engine was introduced into the United States, causing a gradual revival of the steam-engine industry.

The cylinder of the unaflow engine is about twice as long as for a counterflow engine of the same power. The piston is also much longer, occupying about 45 per cent of the cylinder volume. There is a steam valve at each end of the cylinder (Fig. 191), and a row of exhaust openings around the middle of the cylinder. The steam enters at the ends of the cylinder, and the piston acts as the exhaust valve. The steam flows toward the middle, during expansion, and at exhaust it passes out through the exhaust ports.

The indicator diagrams in Fig. 192 show clearly that the compression of the steam continues much longer in the stroke than is the case in the counterflow or parallel-flow cylinder. The steam is compressed to practically the steam-line pressure and temperature, thus eliminating the loss due to initial condensation. Expansion and compression

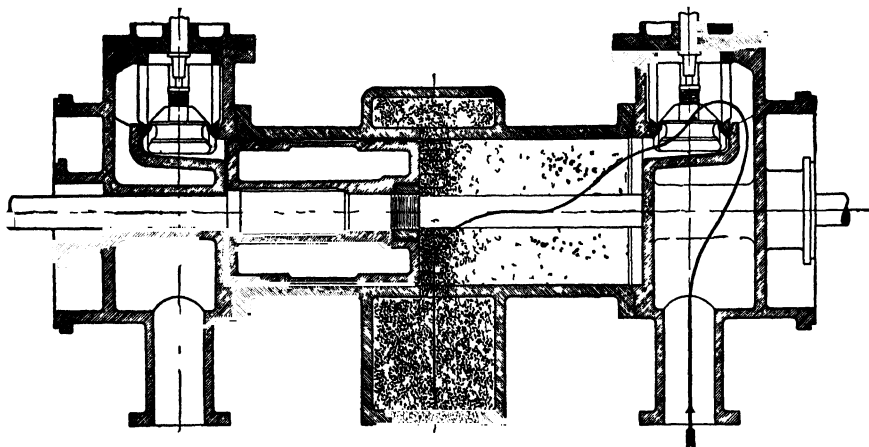
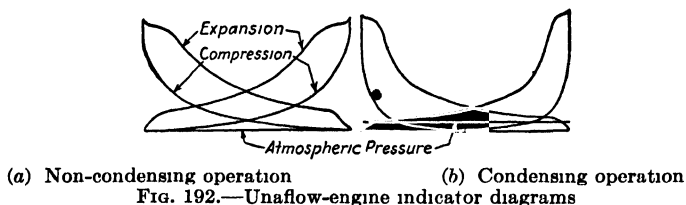



FIG 191.—Section view of a Nordberg unaflow engine cylinder showing the path of the steam

approximate true adiabatic changes of state, resulting in negligible heat loss to the cylinder walls. The main heat loss is that due to incomplete expansion, and this is quite small.

Thus, the unaflow engine reduces to a minimum those losses which had previously been a handicap in reciprocating steam engines. In

(a) Non-condensing operation (b) Condensing operation
FIG. 192.—Unaflow-engine indicator diagrams

sizes up to 500 or even 1,000 hp., its economy compares very favorably with the turbine.

The unaflow engine is especially adapted to condensing operation. It will not give as high steam economy if designed for non-condensing operation, because the non-condensing engine is provided with a larger clearance volume. The purpose of the large clearance volume is to prevent the pressure at the end of compression from exceeding the incoming steam pressure, and it results in a greater amount of

steam being used per stroke. In general, the difference in the economy of a non-condensing unaflow engine and a well-designed counterflow engine is less than in condensing operation.

The unaflow engine, for condensing operation, must have some provision, automatic or manual, to eliminate excessive compression pressure if the exhaust vacuum breaks owing to accident. In such an event, the engine would, of necessity, operate non-condensing. Unless a change is made in the cycle, the top of the indicator diagram will show a large loop of negative work, caused by the compression curve

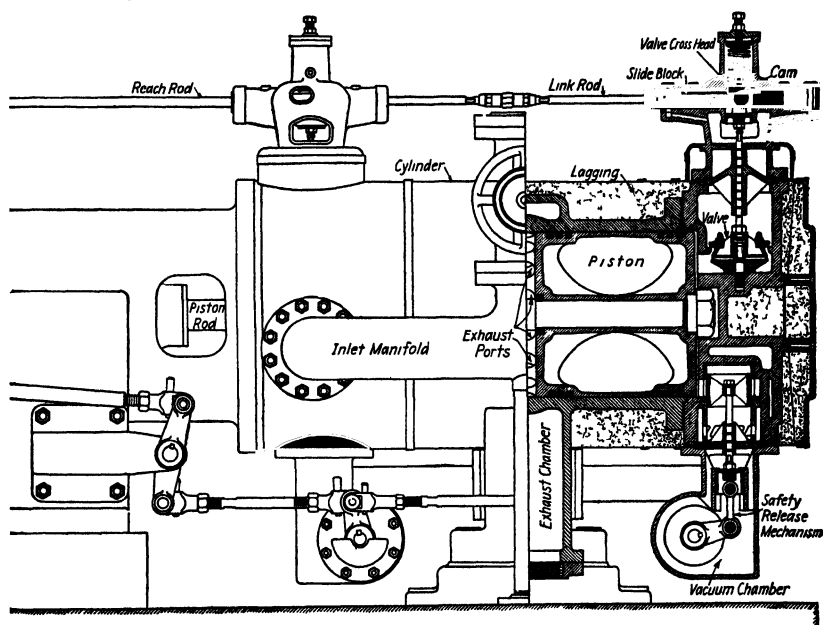



FIG. 193.—Showing cylinder of Murray unaflow engine.

passing above the steam-admission line. If the engine can carry the load, it will do so with a decided jump in steam rate.

The methods used in providing for this change, from condensing to non-condensing operation, or *vice versa,* may be included in two general classes: (1) those which delay the time of the start of compression (delayed compression), and (2) those which increase the clearance volume (increase clearance). The two methods will be illustrated in the following descriptions of commercial types of unaflow engines.

The distinctive features of the different designs of unaflow engines lie mainly in the valve gear used for the steam and auxiliary valves.

214. Murray Unaflow Engine.—This engine is illustrated in Fig. 193. It is built in sizes of from 75 to 650 hp. The cylinder heads

are steam jacketed, this space forming the steam chest, and the cylinder barrel is lagged with an insulating material several inches thick.

The steam valve is of what is called the resilient, double-beat, poppet type. Formed on it is the lower valve seat, while the upper valve seat is a flexible steel disc attached to the valve body by cap screws. The upper valve disc rests on a seat on the head casting, while the lower valve rests on a seat that is a separate casting, held to the cylinder head by a set screw.

The governor eccentric, through an eccentric rod and rocker arm, drives the reach rod. This rod and a link rod are attached to two steel blocks which slide in cavities in each valve bonnet. Each slide block carries a roller which makes contact with a cam placed on a crosshead and fastened to the valve stem. As the slide block receives a reciprocating motion, the roller lifts the cam, thus raising the valve and admitting steam to the cylinder.

The governor is of the Rites inertia type, and controls the timing of the steam valve, thus changing the admission and cut-off only.

The Murray unaflow engine, when designed for condensing operation, is equipped with auxiliary exhaust valves to delay compression. They are used when it is necessary to operate non-condensing. Under such conditions, it becomes a four-valve engine, with an exhaust valve and an inlet valve at each end of the cylinder. The greater part of the exhaust steam, however, passes through the main, central ports. At the start of the exhaust stroke (either end), the auxiliary valve of that end is closed, and does not open until just before the piston closes the central ports. Hence, compression does not start when the piston covers the central ports, as is the case in condensing operation, but exhaust continues through the auxiliary port. When the piston has traveled through approximately three-fourths of the return stroke, the auxiliary valve closes and compression starts. Under this condition, the final compression pressure attains to about that of the incoming steam.

The auxiliary exhaust valves are operated from an eccentric fixed to the shaft. This eccentric, through rods and rocker arm, gives a semi-rotary motion to each auxiliary valve shaft. A crank on each shaft effects a reciprocating motion to the piston valve, thus opening and closing the auxiliary exhaust port at the proper time. When operating condensing, the auxiliary valves are uncoupled.

215. Skinner Unaflow Engine.—This engine is illustrated in Figs. 194 and 195. It has poppet valves of the double-beat type for admission, and the cylinder heads are jacketed with live-steam chambers.

The main feature of this engine is in the automatic, auxiliary exhaust valves, and the automatic method of placing them in action

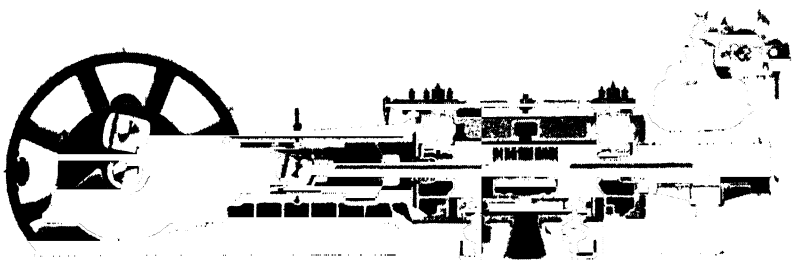



FIG 194 —Large size "Universal" (Skinner) unaflow engine

when the engine suddenly changes from condensing to non-condensing operation. A cross-section of this valve is shown in Fig. 196.

The Skinner unaflow engine may be classed with those that use the *delayed compression* method of eliminating high-compression

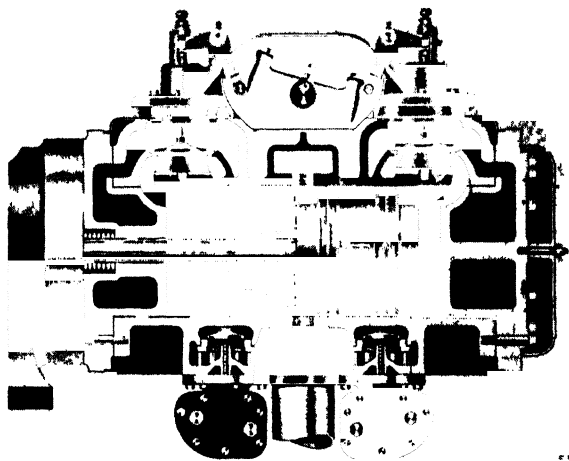



FIG 195 —Cross-section of a small-size Skinner unaflow-engine cylinder showing piston and valves

pressures, while operating with the back pressure at or above the pressure of the atmosphere. The auxiliary valve openings are located at a point about one-third of the return stroke from each end of the cylinder, as shown in Figs. 194 and 195. When operating non-condensing, the cycle of operation is similar to that used by the Murray unaflow engine, operating under the same conditions. The major

portion of the steam exhausts from the main, central ports, and compression does not start until the flow of steam through the auxiliary

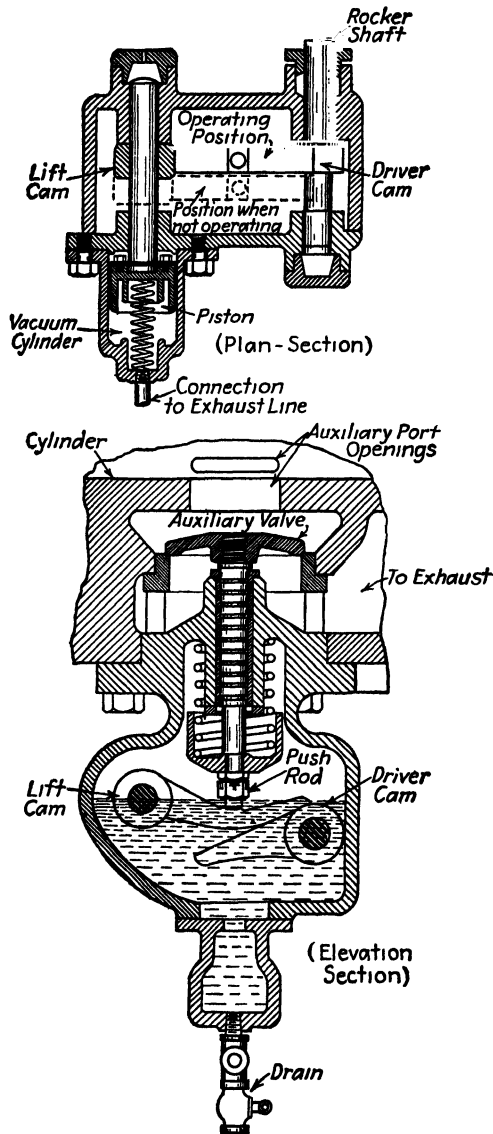



Fig. 196.—Auxiliary exhaust valve of the Skinner unaflow engine.

port ceases. In the Skinner design, the piston,. during the return stroke, stops the steam flow through the auxiliary valve, but it is not closed until after the piston has completely covered the port.

Thus, the auxiliary valve both opens and closes when the pressures on both sides of the valve are very small and practically equal.

The auxiliary valve gear is driven from a fixed eccentric mounted on the crank shaft, and it is in operation whenever the engine is operating. The device illustrated in Fig. 196 is automatically operated, and is meant to prevent the auxiliary valve from operating when there is a vacuum in the exhaust pipe. It also brings this valve into action if the back pressure rises to the pressure of the atmosphere, or higher.

As shown in Fig. 196, there are two cams in an oil bath. The lower or driver cam receives a rocker motion from the eccentric, through the eccentric rod and rocker arm. In order to actuate the valve, the lift or idler cam must engage with the driver. At the end of the idler

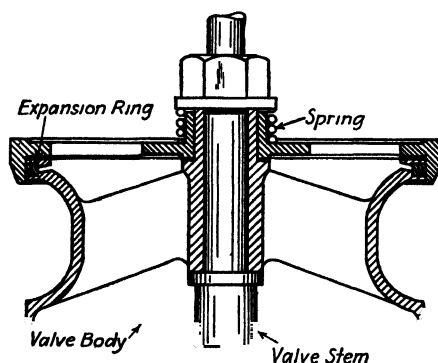



FIG 197 —Section view of Skinner unaflow poppet valve

cam shaft is a small vacuum cylinder connected by a small pipe to the main exhaust. Attached to the idler cam shaft, and in the vacuum cylinder, is a close-fitting piston which is held in operating position by a coil spring. In the operating position, the valve is actuated by the driven cam, the motion of which is predetermined. However, if the vacuum in the main exhaust line rises above that for which the spring is set, its force is overcome, the idler cam is shifted out of engagement with the driver cam, and the valve remains closed, although the driver cam continues moving.

To prevent leakage of steam past the admission valve, when closed, the top part of the valve is made separate from the main part of the body and is held down by a coil spring (Fig. 197). The expansion ring acts as a seal to prevent the leakage. When any unequality of expansion between the valve and its seats occurs, it is taken care of by a slight displacement of the upper part of the valve, with respect to the valve body.

The cam rocker receives a semirotary motion from the governor-controlled eccentric. On each end of the rocker is a case-hardened, removable, steel cam. These cams make contact with a roller on the bell crank, turning the bell crank and opening the valve.

On large Skinner unaflow engines (Fig. 194), the full-floating piston design is used. The weight of the piston is supported by a crosshead at each end of the cylinder. A compensating valve gear allows for unequal expansion of the cylinder and the main valve gear. The inlet valves are operated by cam rockers mounted on a shaft that is parallel to the axis of the cylinder, and is driven, by a linkage, from the governor eccentric. The faces of the valve cams are wide enough to permit the valve-lifting rollers to slide along the face of the cam when the cylinder expands and lengthens.

216. Ames Unaflow Engine.—This engine is similar to the Murray engine already described. The Ames unaflow engine may be furnished with two pistons; a flat-faced one for condensing operation, and a concaved piston for non-condensing operation. The concaved piston provides the increased clearance necessary to prevent high-compression pressures. This type of engine is adaptable for service where exhaust steam is required for heating, and during the warmer season, it may be operated condensing, with higher efficiency.

217. Compression Control.—In addition to that mentioned above, two other methods are used for attaining both condensing and non-condensing operation. The four-valve, controlled-compression engine uses the *delayed-compression* method. The auxiliary valve gear is manually engaged to operate when the engine is operating non-condensing. The exhaust valve is at the end of the cylinder, and is of the balanced design, similar to the steam-inlet valve. The bottom of the exhaust valve stem is attached to a sliding crosshead, carrying a roller, and the roller rides on a cam which has a semirotary motion. The cam is designed to give the correct lift to the valve, and the time of closing the valve is adjustable to suit the back pressure.

The *increased-clearance* design of compression control employs an automatic by-pass valve for controlling the opening of the clearance pocket in each end of the cylinder. An illustration of the Ames type of by-pass valve is shown in Fig. 198. It has a mushroom poppet valve, with the bottom attached to a sylphon bellows. A small pipe connects the space under the sylphon to the main exhaust line. An adjustable coil spring, as shown, is located in this chamber for exerting an upward pressure on the valve stem. When operating condensing, the vacuum causes the sylphon to be collapsed. This compresses the spring, and holds the valve firmly on its seat. If the vacuum

ceases, the spring opens the valve, and, in so doing, opens the clearance pocket to the cylinder, and thereby increases the clearance volume. By means of the adjusting screw, the spring tension can be changed, and, in this way, the pressure at which the valve opens may be varied.

218. Elliott Unaflow Engine.—The Elliott engine is built for condensing service, has double-beat, poppet, steam valves, and a straight-line valve gear, similar to that used on the Murray unaflow engine.

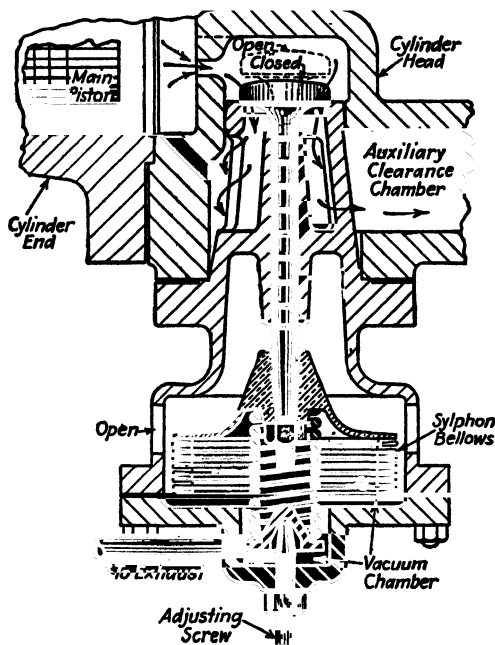



FIG. 198.—By-pass valve of the Ames unaflow engine.

The non-condensing Elliott unaflow engine is similar to the Murray (Fig. 193), being equipped with four valves. The condensing Elliott engine is similar to the Skinner in provision for non-condensing operation.

The combination relief valve (Fig. 199) operates as follows: The small inside valve is to take care of cylinder drain, and the next larger is a spring-loaded relief valve. The latter valve is opened by excessive cylinder compression pressure, and it connects the cylinder with the exhaust pipe. The large outside valve is hand operated and opens the port between the cylinder and clearance pocket for non-condensing operation.

Should the vacuum at any time fail, the spring-loaded valve will relieve excessive compression pressure from the cylinder. For most

economical non-condensing operation, the large valve must be opened, giving the cylinder the increased clearance required.

219. Nordberg Unaflow Engine.—The cylinder of the Nordberg unaflow engine is shown in Fig. 191. The engine uses a lay shaft along the side, parallel to the axis of the cylinder and driven from the crank shaft, through bevel gears. The auxiliary and exhaust valve eccentrics are mounted on this lay shaft and are connected to the valve-

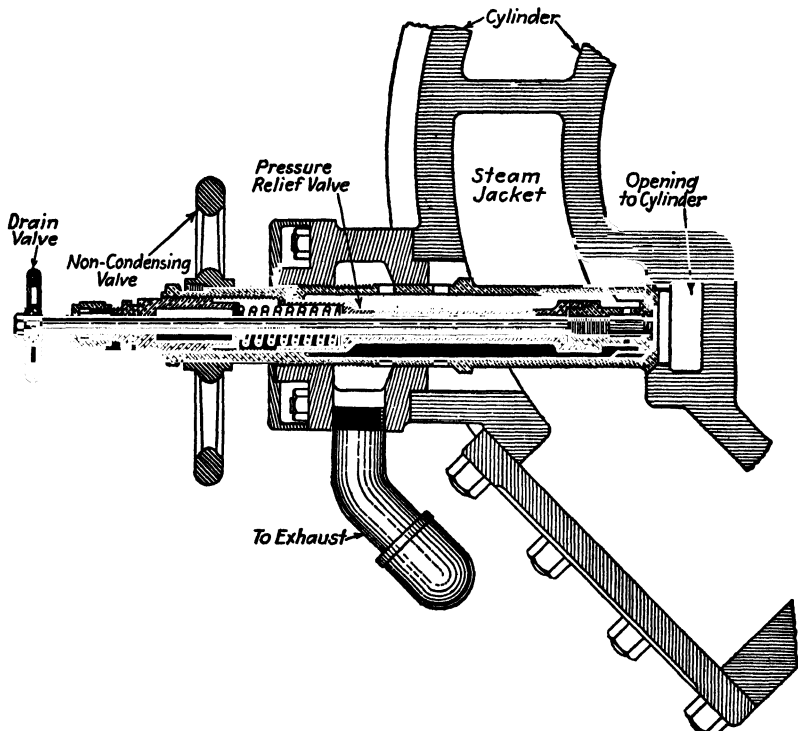



FIG. 199.—Cylinder relief valve of the Elliott condensing unaflow engine.

operating mechanism by rods. A centrifugal governor is supported on the lay shaft, and, being connected to the steam eccentric by a sleeve, controls the engine speed. This is done by changing the eccentric travel and the lift of the steam valve. The valve is of the double-beat, poppet type and is placed in a cage made of metal from the same heat. Therefore, the rates of expansion of the valve and its seat are identical, thus preventing leaks.

A positively operated, auxiliary exhaust valve is placed at each end of the cylinder for use when operating *non-condensing*. A quadrant lever is provided so that in one extreme position the exhaust

eccentrics impart no motion to the valves. In the other extreme, full motion is imparted to them. In intermediate positions of the

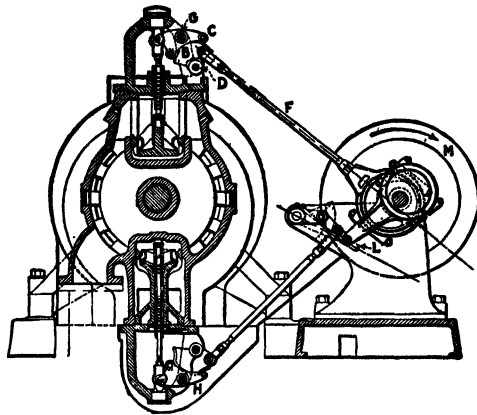



Fig. 200.—Sectional view showing valve gear of the Nordberg unaflow engine.

lever, the time of the start of compression can be varied, and this adjustment can be made when the engine is in operation.

The operation of the steam and exhaust valves, in the Nordberg unaflow engine, is similar. A cross-section through the cylinder, showing these valves, is shown in Fig. 200. As the eccentric rod moves upward, the cam *B* is rotated inwardly. It makes contact with the opening roller. At the same time, the back and depressed part of the cam *B* comes under the closing roller *C*. Further upward motion of the eccentric rod *F* causes the cam to force the opening roller upward, moving about fulcrum *G*. The valve is lifted from its seat. The eccentric passes its extreme position and rod *F* reverses its travel. Moving downward, it causes cam *B* to strike the roller *C*, and so closes the valve. The opening and closing of the valve are rapid.

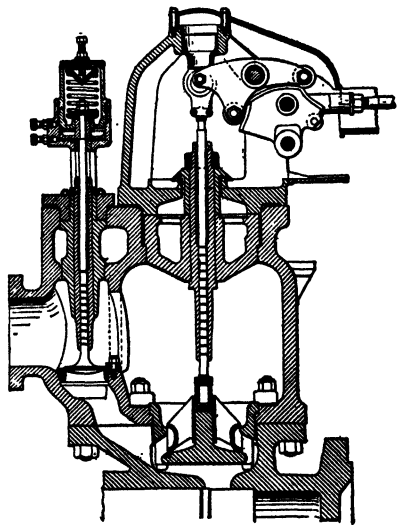



Fig. 201.—Sectional view showing the steam and relief valve of a large Nordberg unaflow engine.

An interesting unaflow installation is a four-cylinder Nordberg unaflow engine used in a continuous steel mill. At 45 per cent cut-off, and

75 r.p.m., this engine will develop 14,000 i.hp. Also, a reversing mill engine at 150 r.p.m., and maximum cut-off, will deliver 30,000 i.hp. This amount of power is, of course, only momentary as a reversing engine may be reversed from full forward to full backward speed as often as twenty times a minute. Each engine has four cylinders, 36 in. diameter by 60 in. stroke, and operates on steam at 240-lb. gage and 125°F. superheat.

The steam valve and gear of the large Nordberg engine is like that just described, but, in addition, a spring-loaded relief valve (Fig. 201) is provided at the main steam admission. Should the pressure in the cylinder rise beyond that in the steam chest, the relief valve opens, allowing the steam to flow back into the steam pipe.

An outstanding installation is the Phillip Carey plant in Cincinnati, Ohio. The prime movers are two 3,750-kw., triple-expansion, vertical, German-made engines. Operating at 225 r.p.m., and receiving steam at 1500 lb. per square inch pressure and 800°F., they exhaust to evaporators at 65 lb. per square inch pressure. Each engine has two single-acting, 14½-in. diameter, high-pressure cylinders, two single-acting, 21¼-in. diameter, intermediate-pressure cylinders, and one double-acting 25⅝-in. diameter, low-pressure cylinder. The two high-pressure cylinders are above the intermediate-pressure cylinders, and the high-pressure pistons are in tandem with and opposed by the intermediate-pressure pistons. Each engine has only three cranks.

Live-steam reheaters are located beside the engine and heat the steam leaving the intermediate-pressure cylinders to 526°F. Piston-type valves are used.

220. Economy of Unaflow Engines.—*Steam economy* in unaflow engines is obtained chiefly in two ways: (1) by steam jacketing the cylinder heads, and (2) by the one-directional flow of steam. By these means, practically all the heat loss due to initial condensation of steam, one of the most serious losses of the counterflow engines, is eliminated. Also, the wire drawing (or throttling of high-pressure steam) at cut-off, which reduces the available energy of the entering steam, is greatly reduced by the quick-acting poppet valves. A minimum of friction loss in the operation of the valve gear is obtained by using balanced poppet valves. There is no sliding action, and a slight lift of the valve gives full opening for admission of steam. As a result of the reduction of these serious losses, met with in the usual counterflow engine, the unaflow engine will develop the same power on less steam. Figure 202 shows comparative steam-consumption curves from different types of engines of about the same capacity. It should be noted that not only does the unaflow engine show the

lowest consumption, but its performance curves are flatter, showing nearly as good economy for light loads as for full load.

The unaflow engine is especially adapted to comparatively high rotative speeds and high steam pressures. It can operate to advantage, using superheated steam as the poppet valves are not troubled by warping, and as they have no sliding action, do not have to be lubricated at points in contact with steam.

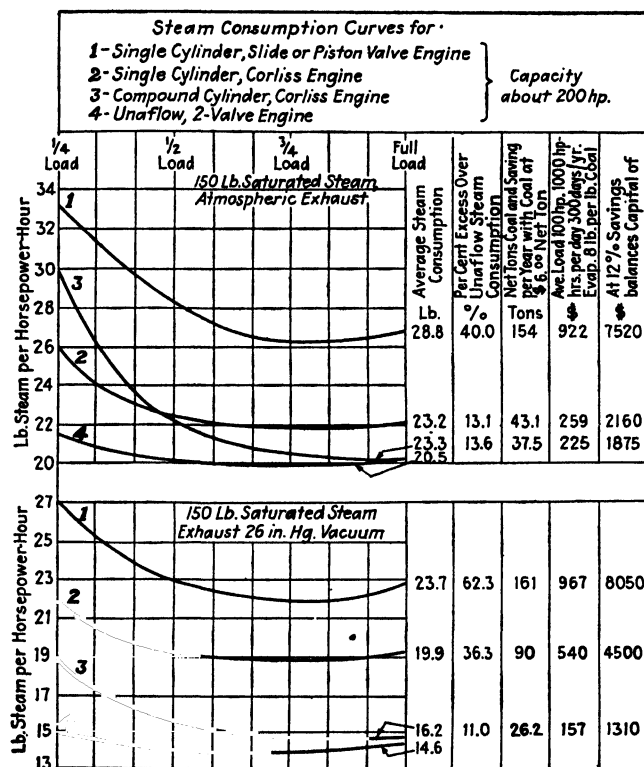



FIG. 202.—Comparative steam-consumption curves.

In small-sized power units, the unaflow engine competes favorably with small turbines.

221. Compound Steam Engines.—Compound steam engines expand the steam, in steps, in separate cylinders. The principal gain in so doing is the reduction of the loss due to initial cylinder condensation. The steam in the high-pressure cylinder of the compound engine expands to an intermediate pressure which is considerably higher than that of the engine exhaust. The temperature of the steam as it leaves this cylinder is, therefore, considerably higher than for the com-

plete expansion, and the cooling of the walls is much less. Successive expansions occur in other cylinders. The temperature range of operation for each cylinder is less, and this enables the compound engine to show better steam economy than the single-expansion steam engine.

Compound engines are classified as follows:

1. Tandem compound.
2. Cross-compound.
3. Angle compound.
4. Duplex compound.

Of the above classes the tandem and cross-compound types are the most common. An engine which expands the steam in three successive

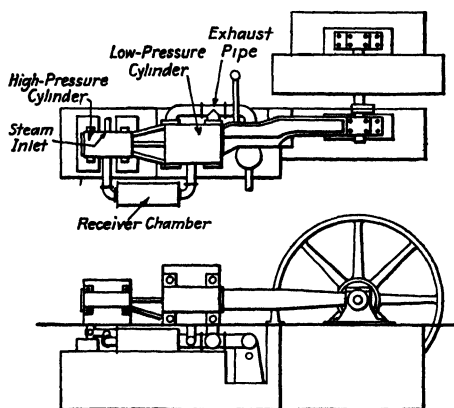



FIG. 203.—Showing arrangement of cylinders in a tandem-compound engine.

cylinders is called a *triple-expansion engine;* and in four successive cylinders a *quadruple-expansion engine.* The name depends on the number of stages and not on the number of cylinders. For instance, a triple-expansion engine might have the exhaust from the intermediate pressure cylinder split, by flowing to two low-pressure cylinders. This engine would have four cylinders, one high pressure, one intermediate, and two low pressure, but would be a triple-expansion engine.

In the *tandem-compound engine,* the pistons are on the same piston rod, and there is only one crank on the engine shaft. The general arrangement is shown in Fig. 203. It is of cheaper construction than the cross-compound engine and occupies less floor space. It does not, however, give as uniform a turning force on the crank shaft.

Without an intermediate receiver chamber, the exhaust from the head end of the high-pressure cylinder enters the crank end of the low-pressure cylinder. The crank-end, high-pressure cylinder exhausts into the head end of the low-pressure cylinder. The steam is admitted

to the low-pressure cylinder during the entire stroke, this being necessary to take care of the high-pressure exhaust. If a receiver chamber is used, earlier cut-off may be provided in the low-pressure cylinder.

In the *cross-compound engine* (Fig. 204) the cylinders are set side by side, each cylinder having a complete operating mechanism. The cranks are usually placed 90 deg. apart, which gives a more uniform turning force on the shaft, and necessitates the use of an intermediate receiver. A high-pressure steam line is commonly connected to the valve chest of the low-pressure cylinder. This permits the engine to be started when the high-pressure piston happens to have stopped on dead center.

FIG. 204.—Showing arrangement of cylinders in cross-compound engine.

The *angle-compound engine* has a horizontal high-pressure cylinder, receiver (steam), and a vertical low-pressure cylinder. Each cyinder has reciprocating parts, but the cranks are both connected to a common crank pin on the main crank shaft.

The *duplex-compound engine* has the low-pressure cylinder directly above the high-pressure cylinder, and the two piston rods are attached to the same crosshead.

Each cylinder of a compound engine has its separate valve gear, and the governing may be effected by one of two methods: (1) by varying the cut-off in the high-pressure cylinder, and (2) by varying the cut-off in both the high- and low-pressure cylinders. The first affects the power of the engine, while the latter method changes only the relative work of the two cylinders. For the best results, the power output of the two cylinders should be equal.

It is generally considered that compounding improves the steam economy by from 10 to 25 per cent, for non-condensing engines, and by from 15 to 40 per cent, for condensing engines.

222. Special Calculations for Compound Steam Engines.—Methods for the calculation of certain characteristic data concerning compound steam engines are given in the following.

The *number of expansions*, or the *ratio of expansion*, as it is sometimes called, is found by obtaining the product of the number of expansions in each successive cylinder; or, it may be expressed as the ratio of the volume of the low-pressure cylinder (or cylinders, if more than

one) to the volume of the high-pressure cylinder at the point of cut-off. The number of expansions for a double-expansion compound engine may be found by the use of Eq. (134), which is derived according to the following:

Consider the clearance volume to be zero, and let

D = the diameter of the low-pressure cylinder, in.
d = the diameter of the high-pressure cylinder, in.
L = the length of stroke, in.
r = the ratio of expansion, high-pressure cylinder.

Using these symbols, the volume of high-pressure cylinder at cut-off may be expressed as

$$\frac{1}{r} \times L \times \frac{\pi d^2}{4}$$

The volume of the low-pressure cylinder may be expressed as

$$L \times \frac{\pi D^2}{4}$$

Then, the number of expansions may be expressed as follows:

$$R = \frac{L \times \pi D^2/4}{1/r \times L \times \pi d^2/4} = \frac{D^2}{d^2} \times r \tag{134}$$

In order to determine the number of expansions accurately, the volume of clearance must be included. R, in this case, equals the ratio of the low-pressure piston displacement volume, plus the clearance volume, to the volume of the high-pressure cylinder at cut-off, including the clearance, also.

The *probable mean effective pressure* for one end of the two cylinders (combined) of a double-expansion engine is expressed by the following equation:

$$\text{Probable m.e.p.} = \left(\frac{p_1(1 + \log_e R)}{R} - p_2\right)f \tag{135}$$

in which

p_1 = initial pressure in high-pressure cylinder, lb. per square inch absolute.
R = number of expansions.
p_2 = exhaust pressure of low-pressure cylinder, lb. per square inch absolute.
f = diagram factor (varies from 0.55 to 0.85).

The *probable indicated horsepower* of the cylinders of a compound engine may be determined by the ordinary formula, using for the area

of the piston, the net area of the piston of the low-pressure cylinder. Theoretically, the power developed by a single-cylinder engine, with cylinder dimensions the same as for the low-pressure cylinder of a compound engine (other conditions being equal) would be the same as that of the compound engine.

The *actual indicated horsepower* of a multiple-expansion engine is equal to the sum of the indicated horsepowers of all of the cylinders.

CHAPTER XII

STEAM TURBINES

223. Introductory.—The steam turbine is essentially a steam engine in which jets of steam flowing within an enclosed chamber at high linear velocity act or react on moving blades mounted on a shaft or drum which is free to rotate. By this means, a part of the kinetic energy possessed by the jets is absorbed by the blades, causing the rotor to revolve. The machine driven by the turbine may be connected to it directly, or through speed-reduction gears.

The turbine has a variety of uses. In large sizes, it is used for driving electric generators and ship propellers. In smaller sizes, it is adaptable for driving pumps, fans, compressors, etc. In general, the turbine is well adapted for work which requires high, rotative, and constant speeds, even with widely fluctuating loads.

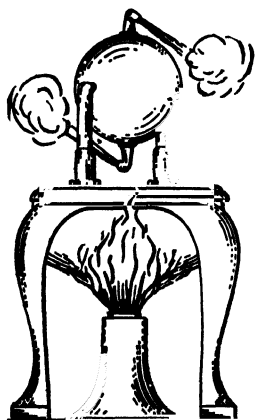



FIG. 205.—Hero's turbine.

224. Historical Notes.—Records relating to the expansion of steam in a rotating engine lead back into the distant past. In the writings of Hero (200 B.C.), a reaction type of turbine is described. This turbine is illustrated in Fig. 205. In reality, it consisted of a hollow sphere mounted on trunnions, with two jet tubes attached on its periphery, as shown. The steam used by the turbine was generated in a boiler below, and it entered the sphere through the hollow trunnions. It was then discharged from the jets, as shown in the figure. These jets were directed tangentially so that the reaction, or kick-back, turned the sphere about the axis of the trunnions.

Following Hero's turbine, there were other inventions of the steam turbine. A prominent example, many years later was Brancas' turbine, in 1629, but no practical use was made of the turbine until near the end of the nineteenth century.

Dr. Gustaf De Laval, a Swedish engineer, invented a cream separator, and in developing a suitable drive for this piece of equipment, experimented with a steam turbine. His first patent was taken out in 1883, and it was for a reaction type of turbine. Later, he

abandoned this type and adopted the simple-impulse turbine, and, in 1889, was granted a patent covering many valuable features. Some of these are still used in modern turbines of the simple-impulse design.

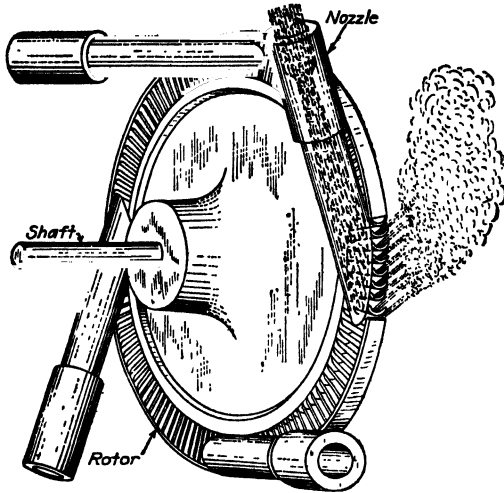



Fig. 206.—Showing nozzles and rotor of a De Laval impulse turbine.

An idea of the difficulties surmounted by De Laval may be had when it is realized that his turbine ran at a rotative speed of about 20,000 r.p.m. and this speed was geared down to about 2,000 r.p.m. for the operation

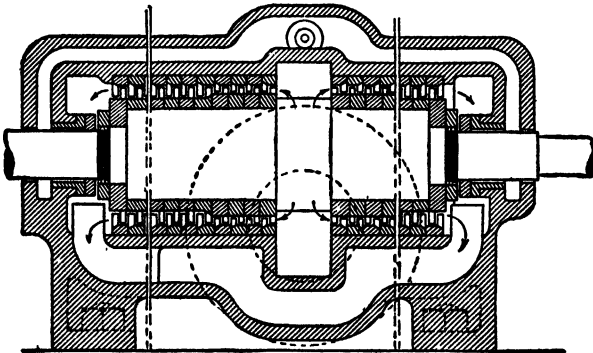



Fig. 207.—Early Parsons turbine.

of the separator. At that time, speeds of about 100 r.p.m. were considered the limit at which gears could be safely used.

De Laval perfected gears with helical teeth. With a large number of these teeth in contact, simultaneously, it was possible to effect the desired reduction in speed without vibration and excessive wear.

Other important features which he invented or developed were the diverging nozzle, rotor disc with symmetrical blades (Fig. 206), flexible shaft with spherical seated bearings, discs of equal strength, discs without hub, bulb-and-shank bracket and throttling governor, all of which are still used on commercial turbines.

At about the same time C. A. Parsons, of England, working independently, developed the reaction turbine, having in view its use on ocean-going ships. In 1884 he took out a patent on a reaction turbine designed to receive steam at the middle of a rotor and exhaust it at each end, after passing it through alternate rows of moving and stationary blades. An illustration of this turbine is shown in Fig. 207. By dividing the flow, the end thrust on the rotor was balanced. The steam expanded in the moving blades, and, on discharging, caused a reaction which impelled the motion of the rotor. Parson's first turbine operated non-condensing, developed 6 hp., used steam at approximately 60 lb. per square inch pressure, and ran at 18,000 r.p.m. The flexible bearing, now in general use, was included in this machine.

In 1888 Parsons obtained a patent on an arrangement of blading whereby the rotor consisted of a cylinder having diameters increasing toward the exhaust end. The blades were mounted on each step, thus taking care of the increasing volume of the steam. At the same time, dummy or thrust pistons to eliminate unbalanced thrust on the rotor were first used. By 1889, 300 Parsons turbines had been built, some as large as 100 hp.

Many engineers added to the early development of the turbine. Among these were Professor Rateau of Paris, France, who, in 1894, designed the pressure-stage impulse turbine, and Curtis, an American, who patented the velocity-stage impulse turbine in 1896.

During 1896 the Westinghouse Electric and Manufacturing Company built the first turbine in this country under the Parsons patent. This was a condensing turbine of 120-kw. capacity; it consumed saturated steam at 150 lb. per square inch pressure, exhausted to a 26½ in. of mercury vacuum and was built to operate at 5,000 r.p.m. In 1899, three 3,600-r.p.m., 300-kw. turbines were installed in the power house of the Westinghouse Air Brake Company, and in 1900, the first central power plant in the United States to be equipped with a steam turbine was that of the Hartford Electric Company, Hartford, Conn. This was a single-flow, 2,000-kw. Westinghouse-Parsons turbine, designed to operate at 1,200 r.p.m. This turbine operated at 155 lb. per square inch gage steam-pressure, 41.6°F. of superheat, and 26.9 in. of mercury exhaust vacuum; water rate, 19.1 lb. per kilowatt-hour.

E. C. Terry, of Hartford, Conn., in 1892, applied for a turbine patent, and by 1903 was producing one turbine of his design per month. Later, in 1907, the first application of turbines for driving centrifugal boiler pumps was made, consisting of eight 300-hp. Terry turbines.

225. Modern Developments.—For electrical generation in central power stations, turbine units of over 100,000 kw. are now not uncommon. Among these are the 104,000-kw. turbine at the Crawford Avenue plant, Chicago, Ill.; the 160,000-kw. turbine at Hell Gate plant, New York City; and the 165,000-kw. turbine at the Philo plant, in eastern Ohio. A turbine unit of 208,000-kw. capacity, consisting of one high-pressure and two low-pressure turbines, is installed and operating in the State Line station, near Hammond, Ind. These turbine units are all of the compound type, having two or three separate cylinders and rotors, and with means of reheating the steam between high-pressure and low-pressure elements.

Other large turbines of interest are the single-shaft, 94,000-kw. tandem-compound turbine, at Long Beach, Calif.; a 160,000-kw., tandem-compound turbogenerator of the New York Edison Company; and, in the Huntley station, Buffalo, N. Y., there is a large single turbine of 75,000-kw. rated capacity.

Steam-turbine design, for central station use, has shown pronounced trends toward (1) high capacity, (2) high steam pressures, (3) high steam temperatures, and (4) use of more efficient plant cycles.

Steam pressures of 1,200 lb. are used at the Lakeside station in Milwaukee, and in the Edgar station at Boston. The Deepwater plant is the first station designed entirely as a 1,200-lb. plant, with two 53,000-kw., cross-compound turbines. There were, in 1935, either in operation or under construction, 10 stations totaling over 300,000 kw. capacity using steam pressures of 700 lb. or over. The Ford plant (Fig. 98), completed in 1936, is the largest 1,400-lb. pressure plant in the world. A large number of stations are operating with steam pressures from 600 to 900 lb.

The vertical-compound or steeple-compound turbine is a new development in high-pressure design. It is arranged to save floor space and reduce the size of building necessary. In this type, the high-pressure turbine and its generator are mounted above the generator of the low-pressure element. At Station A, San Francisco, there are two units of this type. The high-pressure turbines are rated at 12,500 kw., at 3,600 r.p.m., when operating on steam at 1,200 lb. per square inch pressure, 750°F., and exhausting at 450 lb. per square inch. The low-pressure turbine is rated at 37,500 kw. when running at

1,800 r.p.m. and using 750°F. steam at approximately 400 lb. per square inch. The exhaust is at $1\frac{1}{4}$ in. of mercury absolute pressure.

The maximum steam temperature for turbines is 925°F., attained by the Logan plant of the Appalachian Electric Power Company. Several European power stations are using steam at temperatures of from 850 to 900°F.; and as some manufacturers are ready to build equipment to withstand temperatures of 1000°F., there is no doubt but that higher temperatures will be resorted to.

The Detroit Edison Company has been experimenting with the use of steam at a temperature of 1000°F. As a result, they are contemplating the installation of a 10,000-kw., two-cylinder, single-shaft, steam turbine-generator unit to operate on steam having a temperature at admission of 1000°F. and a pressure of 365 lb. per square inch gage. The exhaust is to a pressure of 1 in. of mercury, absolute. The steam, taken from the steam main at 400 lb. per square inch pressure and at a temperature of 725°F., will be passed through an oil-fired superheater designed to superheat 83,000 lb. of steam per hour.

Table 12-1 lists certain stations of recent construction, and gives the principal data concerning their turbines.

The future trend of turbine design may be affected by the experimental work done on the *two-vapor* or *binary-vapor* systems.

Three such units are now in operation with conditions as shown by the following table:

Items of data	Hartford	Kearny	Schenectady
Load on mercury turbine, kw	10,000	20,000	20,000
Speed of mercury turbine	720	900	900
Steam generated, lb. per hour	129,000	325,000	325,000
Steam pressure, lb. gage	275	365	400
Steam temperature, °F	680	750	760
Mercury pressure, lb. gage	70.7	125	125
Mercury temperature, °F	884.5	958	958
Temperature of exhaust mercury	440	485	485
Pressure of exhaust mercury, in. Hg abs	1.5	3	3
Date of placing on line	....	3-25-33	9-25-33
Heat rate, B.t.u. per kilowatt-hour	....	9,500	9,500

It should be noted that the thermal efficiency attained by this cycle is higher than that of the internal-combustion engine. The mercury-steam plant is economically justified where fuel costs are high and where the unit can be given a continuous and constant heavy load. Other two-vapor systems

TABLE 12-1.—CHARACTERISTIC DATA OF VARIOUS STEAM-TURBINE INSTALLATIONS

Company	Location	Date of first operation	Pressure, lb. per sq in.	Temperature, °F.	Main generating units			Bleed points		Remarks
					Number	Type	Capacity, kw.	Number	Pressure	
Appalachian Electric Power Co.	Logan, W. Va.	1937	1,250	925	1	Turbine	40.000	..	....	3,600 r.p.m. turbine
Milwaukee Electric Ry. and Light Co.	Port Washington Station	September, 1935	1,390	840	1	T. C. T.	80,000	5	....	Reheat pressure, 422 lb.
Detroit Edison Co	Conners Creek		710	850	3	C. T.	30,000	4		
City of Fort Collins	Ft. Collins, Colo.	1936	425	710	1	C. T.	1,000			
					2	C. T.	1.500			
Firestone Tire and Rubber Co	Akron, O.	1936	1,400	760	1	B. P. T.	10.000	..	....	Exhaust pressure, 250 lb.
Ford Motor Co	Dearborn, Mich.	1936	1,200	900	1	B. P. T.	15,000	4		
					1	V. B. C. T.	110,000			
Philadelphia Electric Co.	Richmond Sta.	1935	425	850	1	T. C. T.	165,000	4	....	Plant addition
Weirton Steel Co.	Weirton, W. Va.	1936	900	815	1	B. P. T.	10,000	..	....	Exhaust pressure, 215 lb.
Iowa Electric Light and Power Co.	Cedar Rapids, Ia.	July, 1936	750	765	1	B. P. T.	5,000			
Pioneer Flintkote Co.	Vernon, Calif.	August, 1936	450	...	1	B. C. T.	7,500	2	30-125	
Franklin Board and Paper Co.	Franklin, O.	1936	400	600	1	B. C. T.	1,500	3	15-50-150	

C. T. = condensing turbine; T. C. T. = tandem-compound turbine; V. C. T. = vertical-compound turbine; B. P. T. = back-pressure turbine; B. C. T. = bleeder-condensing turbine; V. B. C. T. = vertical-bleeder compound turbine.

which are being studied are: diphenyl-oxide steam, aluminum-bromide steam, and the zinc-ammonium-chloride cycle.

A rather unexpected development was the elimination of the power-house structure to lower the first cost of plant. Two out-of-doors plants are in operation. In the one at Schenectady the 20,000-kw. mercury turbine generator, the 6,000-kw. steam turbine, feedwater heaters, and evaporators are installed out of doors on the turbine floor level, which forms the roof of the building.*

In these plants the mercury vapor, exhausting from the mercury turbine, is passed to a condenser which is essentially a steam boiler. The steam generated in the condenser is later superheated in the furnace of the mercury boiler and then passed through a steam turbine unit. Later developments of this type of power unit are being awaited with much interest.

226. Classification of Steam Turbines.—Turbines are generally classified into the following types:

1. Impulse.
2. Reaction.
3. Combination impulse and reaction.

In an *impulse* design of steam turbine, there is an attempt to attain no expansion of steam within the moving blades. Rather, it is expanded in stationary nozzles in which a high velocity is acquired. The velocity attained in a properly designed nozzle depends on the pressure drop through it. Part of the heat energy of the steam is transformed into kinetic energy, in the nozzle, and most of this energy is imparted to the moving blades by the resulting steam jets. On leaving the blades, in the actual turbine, the steam does have a slight reaction effect.

Impulse turbine designs may be classified as *single stage, multiple-pressure stage, multiple-velocity stage*, and a *combination* of the latter two. The term *pressure stage* refers to a unit of nozzles combined with moving blades in which the energy of the steam, resulting from each pressure drop, is utilized. *Velocity-stage* refers to the number of moving blades (or blade discs) in which the energy of the steam in each pressure drop is absorbed.

Hero's rotating sphere (Fig. 205) is an example of a pure *reaction turbine.* In the so-called reaction turbine of the present day, the pressure drop (expansion of steam) occurs within both the stationary and moving blades. Thus, the rotation of the shaft or drum carrying the blades is the result of both impulse and reactive forces in the steam. One stage of a reaction turbine includes a row of stationary blades

* The above extract is taken from a paper by Dudley P. Craig in *Industrial Power*, vol. 30, No. 6, June, 1936.

and the following row of moving blades. Because of the small pressure drop in each stage, the number of stages required in a reaction turbine is much greater than in an impulse turbine of the same capacity.

The *combination impulse-and-reaction* type of turbine consists, generally, of impulse elements in the first or high-pressure and high-temperature section, and reaction elements in the section where the pressures and temperatures are lower. The purpose of such a combination is to eliminate difficulties of design and construction incident in the expansion of steam at extreme conditions in reaction turbines. It will be seen later in the chapter, that the type of construction used in impulse turbines is more readily adaptable than that of reaction turbines to the utilization of steam at high pressures and temperatures; the chief reason being that, in the former, there is no necessity for small clearance spaces between the stationary and moving elements to be seriously affected by expansion when high-temperature steam is being used. Also, the possibility of excessive and wasteful leakage of steam between the various high-pressure stages (steps of expansion) can be avoided.

The prominent manufacturers of various types of turbines are subdivided, as follows:

1. Impulse, single stage.
   a. De Laval.
2. Impulse, velocity stage.
   a. Curtis (original).
   b. Westinghouse, reentry.
   c. De Laval.
   d. Terry.
   e. Sturtevant.
   f. Elliott.
   g. Allis-Chalmers, impulse.
3. Impulse, pressure stage.
   a. Rateau (French).
   b. De Laval.
   c. Terry.
   d. General Electric (Curtis).
   e. Kerr (Elliott).
   f. Moore.
4. Impulse, compound-pressure, compound-velocity stage.
   a. Moore.
   b. General Electric.
5. Reaction.
   a. Parsons (England).
   b. Allis-Chalmers.
   c. Westinghouse.
   d. Brown-Boveri.
6. Combination impulse and reaction.
   a. Westinghouse.
   b. Brown-Boveri.

227. Impulse-turbine Designs.—The principle on which the simplest form of impulse turbine operates may be studied by referring to Fig. 208 *a*. As shown, the entire pressure drop of the steam is effected in one nozzle, or several nozzles in parallel, and the resultant steam velocity is absorbed in one row of moving blades. Due to the high velocity of the steam jets, the rotor of the simple impulse turbine runs at high rotative speeds.

For electrical power generation, impulse turbines are designed for moderate rotative speeds by being built in types of more than one

stage. A compound-velocity turbine (Fig. 208 *b*) has the entire pressure drop in one nozzle, as before, but the steam velocity is absorbed in two or more rows of moving blades. Between the rows of moving blades are rows of stationary blades which serve to direct the steam into the subsequent row of moving blades, at the proper angle.

In a pressure-stage, or compound-pressure-stage, turbine, the pressure drop is divided into a series of smaller pressure drops, as illustrated in Fig. 208 *c*. Steam enters the first nozzle and expands to a lower pressure, thereby acquiring kinetic energy which is imparted

(*a*) One pressure stage.
(*b*) One pressure stage with two velocity stages.
(*c*) Three pressure stages.
(*d*) Two pressure stages, each having two velocity stages.

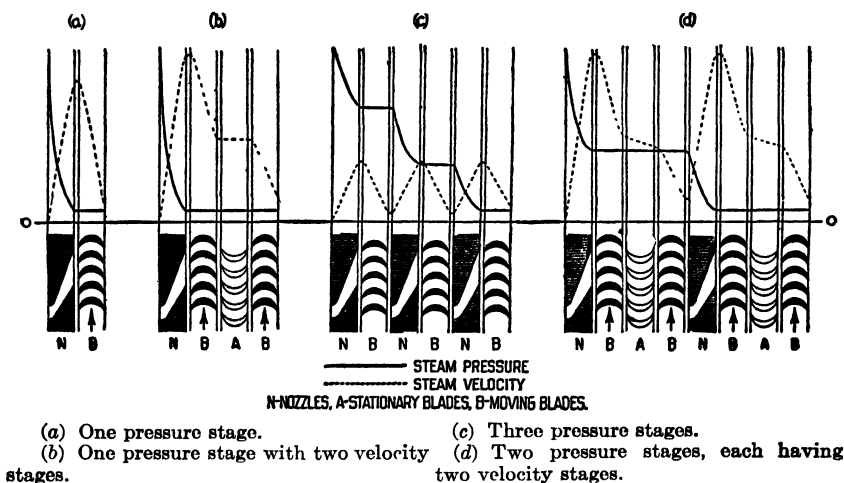



FIG. 208.—Pressure and velocity diagrams, impulse turbines.

to one row of moving blades. The steam then enters the nozzles of the second stage and expands further, regaining velocity; then, it impinges on the second row of moving blades. This operation is repeated in each stage, until, in the last stage, the steam expands to the exhaust pressure and strikes the last row of moving blades, and then flows to the exhaust line.

The compound-pressure, velocity-stage turbine (Fig. 208 *d*) is a combination of pressure and velocity staging. The main purpose of such construction is to effect increased efficiency and, also, to reduce the rotative speed of the shaft. This results from absorbing the energy of the steam in a large number of rows of blading.

228. Single-stage Impulse Turbine.—Turbines of this type have but one rotor disc (Fig. 206) on the periphery of which are attached the blades or buckets. One or several diverging nozzles are used for expanding the steam and discharging it onto the blades, and a governor, driven from the shaft, is provided for regulating the steam flow.

The De Laval single-stage turbine is one of the best known turbines of this type. As built at the present time, it is of simple design, and incorporates many of the features used in De Laval's original turbine.

On account of the high steam velocities occurring in single-pressure-stage turbines, the linear speed of the blades is generally 1,000 to 1,500 ft. per second. At this speed, the centrifugal forces tending to rupture the rotor disc are very great. Experience in building turbines has revealed that a hole through the center of a flat rotor reduces its strength during rotation by approximately 40 per cent. For this reason, the high-speed rotor disc is made solid, with heavy hubs which provide for bolting on of the shaft. A groove is cut on each side of the rotor rim, just inside of where the blades are attached, thus placing the weakest part of the rotor at the rim. Made in this way, a rupture of the rotor will result in the least damage.

Fig. 209.—De Laval blades, showing method of attachment to rim of rotor disc.

Aside from designing a disc with sufficient strength, the fact that it is impossible to make a perfectly true rotor wheel (one in which the geometrical axis coincides with the gravity axis) must be taken into consideration. To eliminate excessive vibration due to this imperfection, a flexible shaft is used, which permits the disc to rotate around its gravity axis. With this provision, the small turbine is safe to operate at speeds as given in Table 12-2. After the turbine, on increasing its rate of rotation, passes its critical speed (about one-fifth to one-eighth of the normal speed), the shaft flexes sufficiently to allow the disc to assume its center of gravity, or gravity axis of rotation. The rotor speed is reduced to a usable speed by special reduction gears having helical or herringbone teeth. The common reduction, in such cases, is 10:1.

The De Laval impulse-turbine blades (Fig. 209) are made of drop-forged steel, and they have shanks especially made with enlarged ends. The blades are mounted, as shown, with the shanks fitting in slots in the rim of the wheel and anchored by the enlargement which fits the cylindrical hole at the bottom. The upper end of the blades have flat extensions which form a support for the outer end of the blade.

The extensions of the adjoining blades fit closely together, and thus form a continuous ring.

TABLE 12-2.—DATA FOR DE LAVAL SINGLE-STAGE TURBINES

Data	Size, horsepower			
	5	30	100	300
R.p.m.	30,000	20,000	13,000	10,000
Blade diameter, in.	3.94	8.86	19.68	29.92
Linear speed of blades, ft. per second	516	772	1,115	1,273

On the outer end of the shaft, a spherical seated bearing is used. This permits the necessary movement when flexure of the shaft occurs. On each side of the gear-reduction pinions are plain babbit bearings.

The objectionable features of the simple impulse turbines are (1) high friction losses in blading, due to the high steam-jet velocity; (2) rotative speeds too high for most purposes; (3) small capacities; (4) difficulties encountered in operating condensing.

229. Multiple-pressure-stage Impulse Turbine.—The design of the first turbine of the pressure-stage type is generally accredited to Professor Rateau (1863–1930), a French engineer.

The pressure-stage turbine is essentially a series of single-stage impulse turbines combined into one unit. The construction is such that the exhaust casing of one stage forms the nozzle plate or nozzle diaphragm of the following stage. This arrangement permits the total pressure drop to take place in a series of distinct steps. The constantly increasing specific volume of the steam, as it approaches the exhaust, is provided for by increasing the sectional area of the space through which it flows. Commonly, the blade length is increased, and the base diameters of the rotor discs decrease, to maintain the mean diameter of the blades constant in all stages. Another method of taking care of the expansion is by increasing the width of the steam belt. In the high-pressure stages, the nozzles occupy only a small arc of the periphery of the rotor. As the steam increases in specific volume, it is necessary to provide more and larger nozzles in the lower stages, and, frequently, they occupy the entire circumference of the interstage diaphragms.

Steam expansion can be carried to very low pressures in turbines of this type; hence, it is well adapted to a wide range of pressures. It is widely used for electrical-power generation, and handles, effectively, the variable loads encountered.

The compound- (multiple) pressure-stage turbine, as compared with the velocity-stage turbine, has the advantage of less friction loss due to the lower velocity of the steam leaving the nozzle. Due to the relatively small pressure drop in each set of nozzles of the pressure-stage turbine, the steam jets have a comparatively small velocity. Also, the steam flow maintains its identity as a jet and is less turbulent than in the velocity-stage turbine.

230. Multiple-velocity-stage Impulse Turbines.—The original Curtis turbine was of the velocity-stage impulse type, and, until about 1910, it was built vertical, with as many as 14 velocity stages. This design was developed by the General Electric Company.

The vertical feature of this turbine embodied a generator above the turbine and the condenser below, giving a very compact design. The vertical shaft was supported on a step thrust bearing containing oil at approximately 900 lb. per square inch pressure. Thus, the shaft actually floated on oil, but on attempting to build large turbines with this type of bearing construction, difficulties arose which led to the horizontal-shaft design.

After building Curtis turbines of the original design for some time, investigation evolved the fact that the steam, after flowing beyond the first four stages, lost its identity as a jet, and whirled and eddied through the casing, accomplishing no useful work. The large number of velocity stages were then dispensed with, and, at the present time, three or four velocity stages are the maximum number used.

Turbines of the velocity-stage type are generally limited in use for driving such equipment as pumps, fans and other small units.

Figure 210 illustrates a sectional view of a two-velocity-stage turbine. It is built thus in capacities of from 1 to 600 hp. A large shaft supports the rotor disc holding the two sets of blades around its periphery, as shown. Each nozzle supports a sector of guide vanes which guide the steam into the second row of moving blades. The nozzles may be closed separately by a flat valve. The centrifugal governor is mounted on the end of the rotor shaft and, through a system of links, operates the balanced inlet valve, thus regulating the speed. On the left of the turbine casing is the emergency governor and automatic stop valve. When the emergency governor operates and releases a trigger, a coil spring pulls downward on a bell crank and pushes the stop valve to the right, shutting off the steam. The steam chambers of the turbine are sealed by three carbon packings on each side of the rotor.

Figure 211 shows several important parts of the *De Laval velocity-stage turbine.* The assembly of the rotor and shaft illustrates the

split bearings and the oil-ring method of lubricating the main bearings On one end of the shaft is mounted the centrifugal governor, as shown, and on the other end is a coupling.

The nozzles of the De Laval turbine are separate castings which are held firmly in fixed positions in openings in the steam chest. The

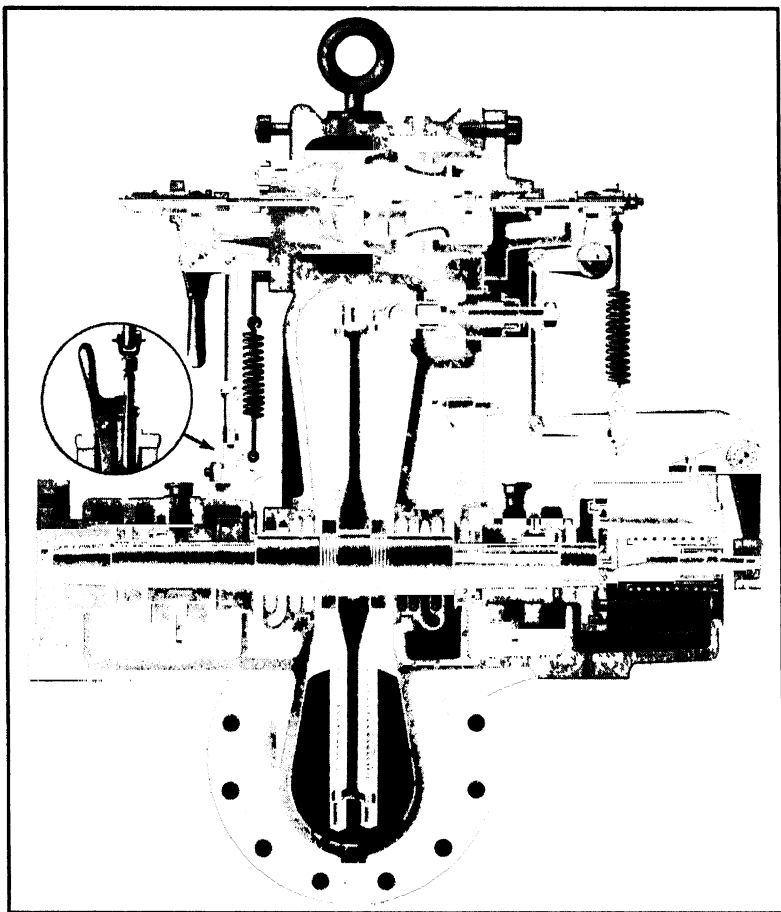



FIG. 210 — De Laval velocity-stage turbine.

nozzle castings support a small sector of guide vanes placed to fit between the two rows of moving blades on the rotor. The stationary blades or guide vanes do not extend around the entire circumference, but just for small sectors opposite each nozzle. Each nozzle is equipped for hand control, in addition to the main governor valve. With the proper control of the number of nozzles in operation, throttling action of the governor can be reduced to a minimum.

The *Elliott turbine* (Fig. 212) is similar in construction to the De Laval turbine. The figure shows the relation of the governor to the throttling valve, and the location of the lubricated bearings outside

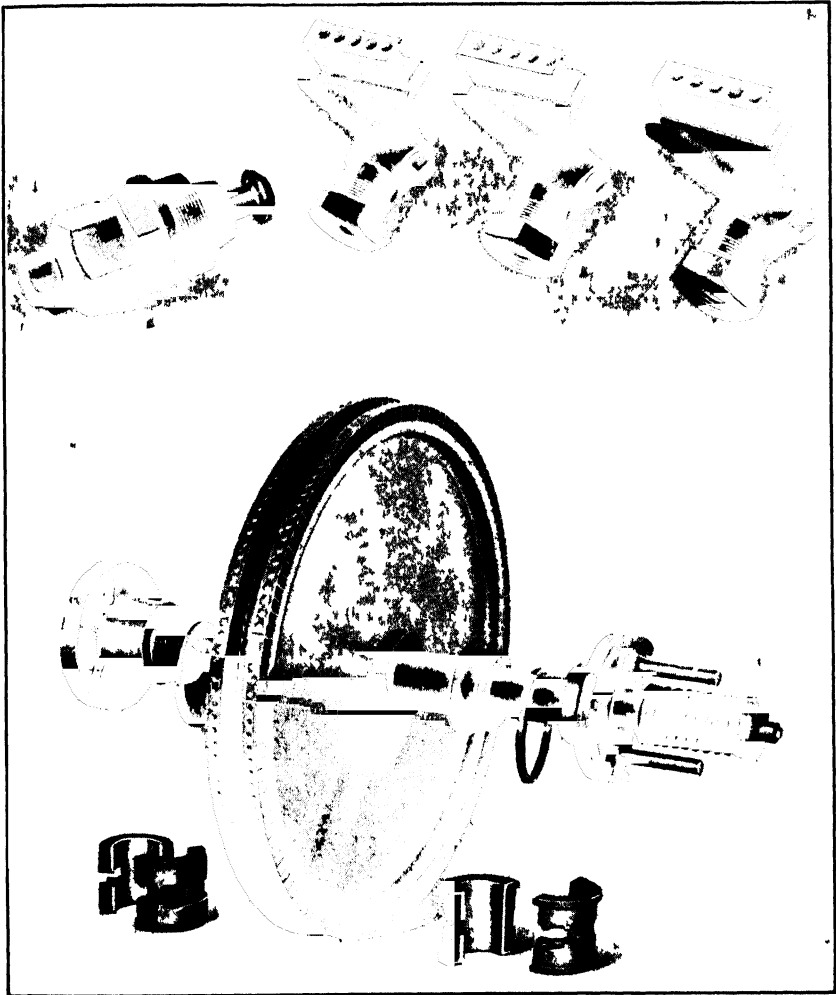



FIG 211 —Showing detail parts of De Laval velocity-stage turbine.

of the shaft packings. Various other important features are shown in the illustration.

The *Terry impulse turbine* is illustrated in Fig. 213. It is in effect, a velocity-stage turbine with one rotor disc. It is different from the axial-flow type in that the steam issues at high velocity from an expanding nozzle and enters the side of the rotor bucket wherein its

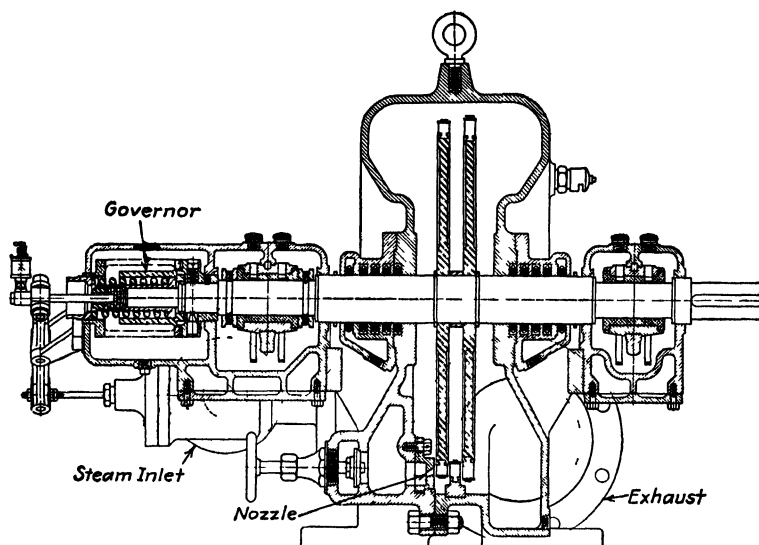



FIG 212 —Elliott velocity-stage impulse turbine

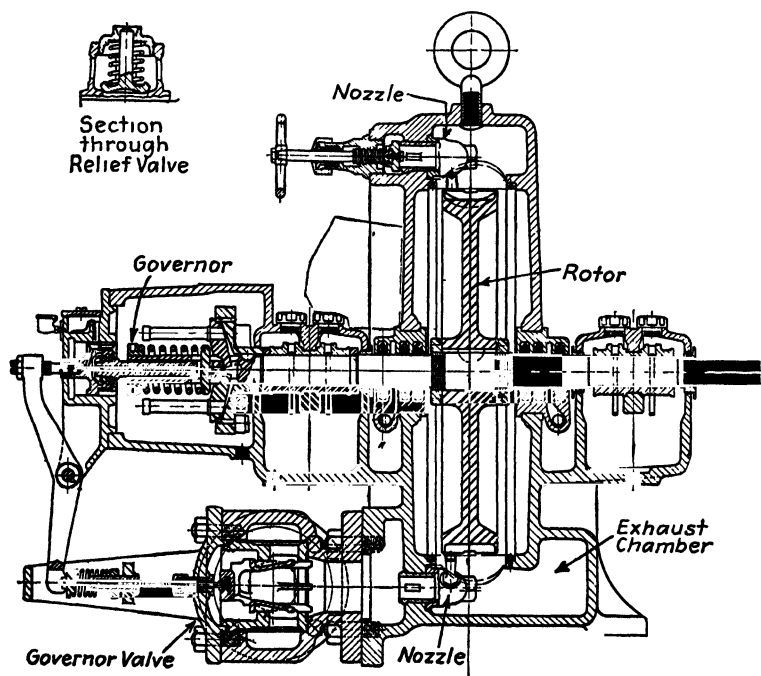



FIG 213 —Section view of Terry impulse turbine

direction of flow is reversed 180 deg. Leaving the bucket, it enters a reversing chamber in the casing and is returned to the wheel. This process is repeated several times, as shown in Fig. 214.

Figure 214 shows the construction of the rotor wheel and also, in phantom view, the action of the steam from one nozzle, thus illustrating very clearly the helical path followed by the steam after leaving the nozzle. The nozzle and reversing chambers, shown in Fig. 215, form a segment which is bolted to the casing. The nozzles are easily

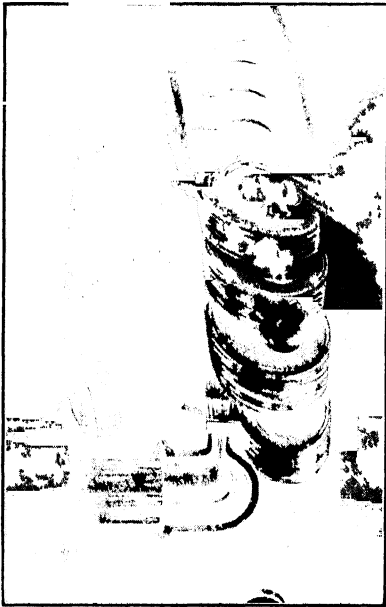



FIG. 214 —Phantom view showing Terry nozzle and buckets.

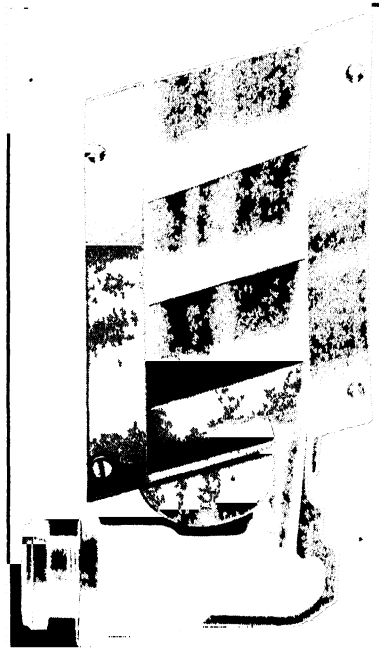



FIG. 215.—Terry nozzle and reversing buckets.

removed and changed to meet new conditions of pressure or capacity. The buckets are semicircular in shape, and are milled out of the solid steel forging. The metal at the middle of each blade is removed to provide a path for eddying steam. This also strengthens the blade by distributing the load along the blade support.

The packing gland used in the Terry turbine consists of three carbon rings, as shown in Fig. 213. Each ring is mounted in a separate casing, and they are permitted to float with the shaft but do not turn. Each ring is composed of three segments held together by a garter spring. Between the two outside rings is provided a drain for the condensed steam leakage. These glands require no lubrication or

cooling and they do not score the shaft. The bearings are split, lined with babbitt, and lubricated by two brass rings.

The *Terry governor,* as shown in Fig. 213, is similar to that used by the smaller De Laval turbine. The centrifugal force of the weights

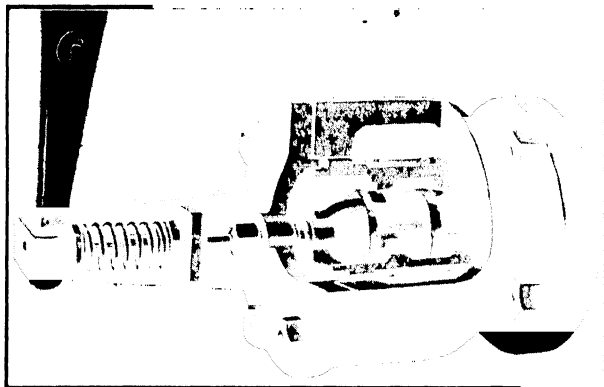



FIG. 216.—Terry governor valve.

is opposed by a compression spring. At the outer end of the spring is an adjusting nut, by means of which small variations in the speed may be obtained. For greater speed ranges, the weights and springs may be changed. In operation, an increase of speed causes an internal

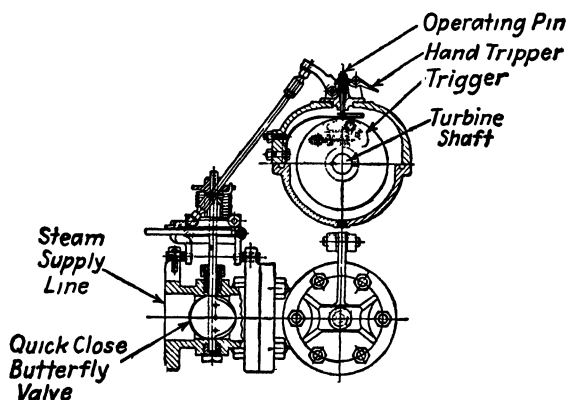



FIG. 217.—Terry overspeed stop.

rod to move outward and, by means of a ball-bearing transmission, acts upon a lever which operates the governor valve. This valve (Fig. 216) is of the double-seated, balanced type. All lost motion is taken up by a spring mounted on the end of the valve stem. The valve cage is equipped with a cylindrical strainer which prevents foreign matter from entering the turbine.

The overspeed stop (Fig. 217) consists of a trigger mounted on the back of the governor disc and held in position by an adjustable coil spring. At speeds above that for which the spring is set, the increased centrifugal force will move the trigger outward, causing it to strike the operating pin. The movement of this pin releases, with a hammer blow, a spring-loaded butterfly valve located in the main steam line, thus shutting off the steam supply. This valve may be tripped by hand if a quick stopping of the turbine is desired.

The Terry turbine (Fig. 213) is used for small drives and is generally operated non-condensing. The turbine casing, including the bearing

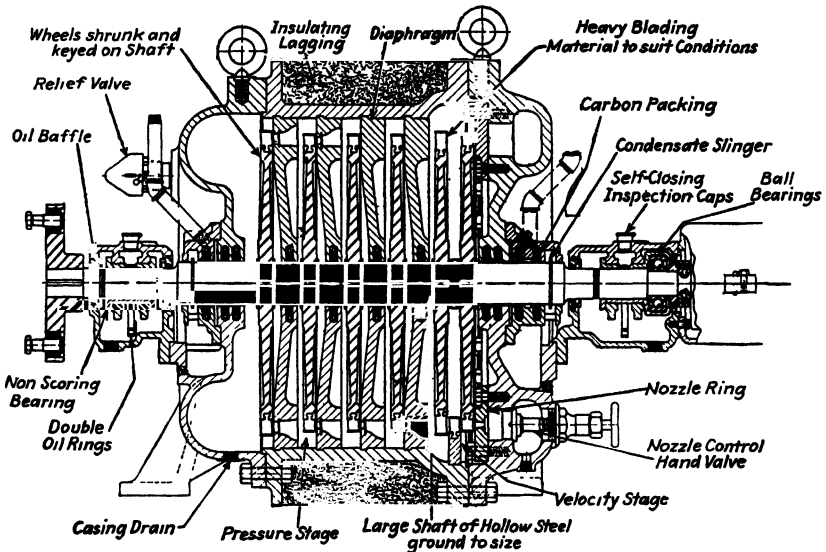



FIG. 218.—Moore pressure-stage impulse turbine.

caps and governor housing, is horizontally split. This permits easy access to the interior for inspection or repair. The steam and exhaust connections are attached to the bottom half of the casing.

A sentinel safety valve is provided on the turbine casing. This operates in case the inlet valve is opened when the exhaust valve is closed. Each nozzle is provided with a hand-operated valve, the number of which in operation at any time should be suitable for the load on the turbine. For light loads a throttling effect of the governor can be eliminated by closing off the desired number of nozzles. For driving slow-speed equipment, the Terry turbine is equipped with herringbone reduction gears.

231. Combined Pressure- and Velocity-stage Impulse Turbines.— Pressure staging, necessitated by increased sizes of turbines, at once

introduced a new field for velocity staging. Turbines of the type to be described in this article may be said to combine many of the advantages of both pressure and velocity staging.

The Moore turbine (Fig. 218) is an example of combined pressure and velocity staging. The first pressure stage includes two velocity stages, those following being straight pressure stages. The rotor wheels are shrunk and keyed onto the shaft. The buckets are held in a dovetail groove in the rim of each wheel and are supported at the

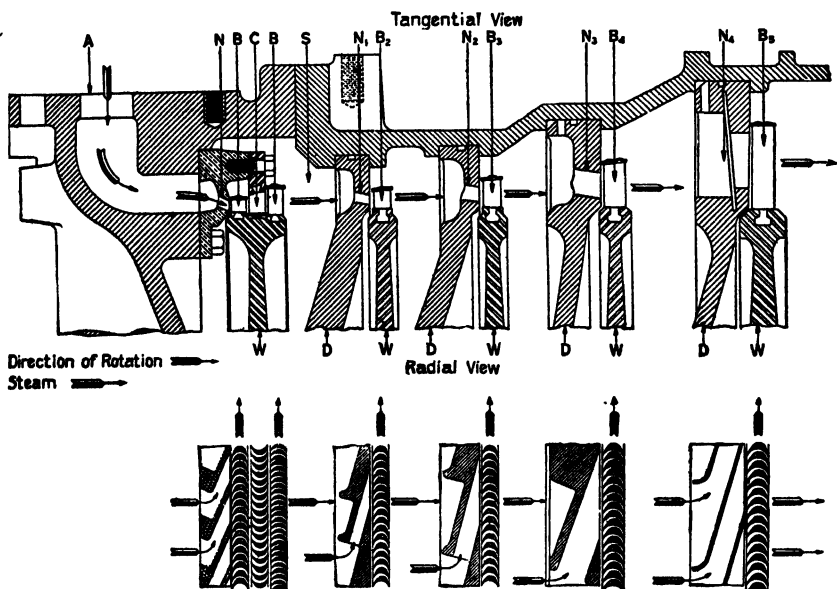



FIG. 219.—Section diagram of a velocity- and pressure-stage impulse turbine. (*General Electric Company.*)

outer ends by a shroud ring riveted to each bucket. Carbon packings are used at the center of each diaphragm and at each end of the casing to prevent leakage. The casing is horizontally split for easy access to the interior. Hand valves are provided for each nozzle for economical operation at light loads.

A sectional view showing the path followed by the steam on flowing through another turbine of the pressure- and velocity-stage type is shown in Fig. 219. The diaphragms D, as shown, divide the inner space of the turbine housing into separate compartments, in each of which is a revolving blade disc W. On the periphery of the discs are mounted the blades or vanes.

After passing through the governor valve, the steam enters the nozzle chamber of the turbine through the port A. It then passes

through the first-stage nozzles N, and in doing so the steam expands to a pressure of about one-tenth of the initial pressure, giving the jet considerable velocity. On leaving the nozzle the steam impinges on the first row of revolving buckets B (Fig. 219), to which it imparts some of its energy. Leaving these buckets it then passes through the row of stationary buckets C, and then onto the second row of buckets, B. Here the steam jet gives up most of its remaining velocity and enters the compartment space indicated by S. In this space, the first stage pressure drop is completed.

From the first stage, the steam enters the second-stage nozzles N_1, where it expands further, thus again acquiring a high velocity before

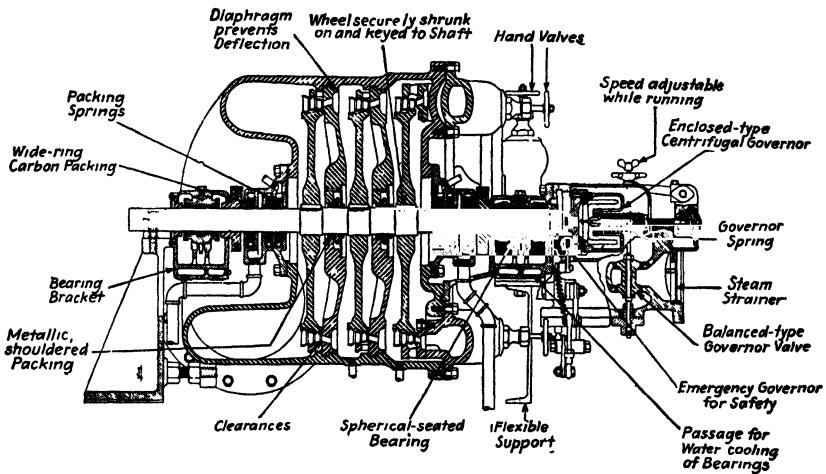



FIG. 220.—Curtis steam turbine Showing compound-pressure and velocity staging. (*General Electric Company.*)

impinging on the buckets B_2. This process is repeated in the subsequent stages until, leaving the buckets of the last stage, the steam is at the exhaust pressure, and all of the available energy of the steam has been used.

As in all impulse turbines, the steam expands only in the stationary nozzles (Fig. 219), and there is no appreciable pressure drop through the blades in any stage. Because of the uniform pressure in all parts of the chamber of each stage, there is very little end thrust on the wheels, and no need for elaborate end-thrust balancing devices. Instead, a small locating thrust bearing is provided to maintain the proper spacing between rotary and stationary parts.

The *Curtis turbine*, having combined or compound-pressure and velocity-stage features, is illustrated in Fig. 220. There are, in this turbine, two velocity stages in each pressure stage. The path followed

by the steam, in expanding and flowing through the turbine, is clearly evident in the figure. Built as shown in Fig. 220, the Curtis turbine units seldom exceed 1,000-kw. capacity. They are, however, built

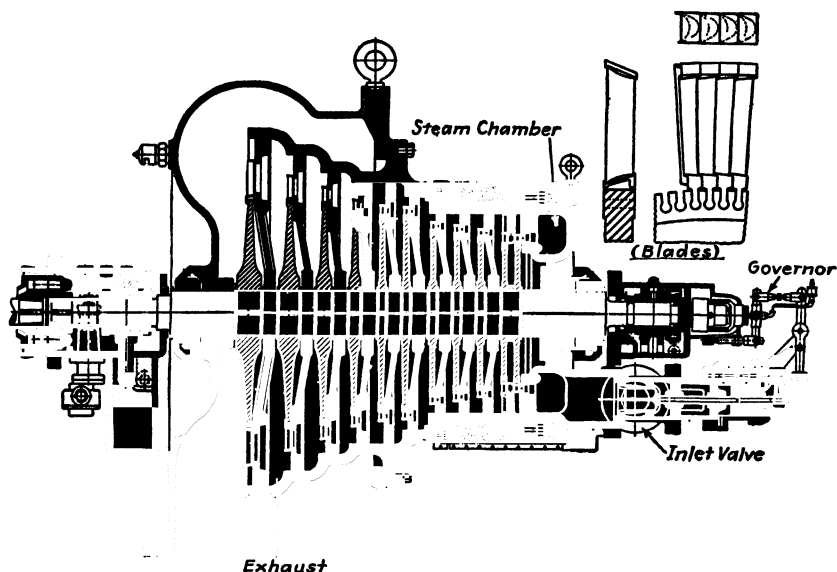



FIG 221 —Kerr impulse turbine.

in the largest sizes of various other designs which are essentially of the straight pressure-stage type.

The Kerr turbine (Fig. 221) has two velocity stages in the first pressure stage, followed by a number of single-pressure stages. Each

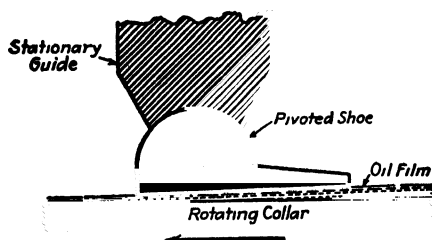



FIG. 222.—Diagrammatic sketch illustrating principle of Kingsbury thrust bearing.

bucket wheel is keyed and shrunk on the shaft, and is held in place, axially, by shrink rings. The buckets are rigidly fastened to the wheel by bulb shanks driven into slots in the wheel rim. Shroud-ring elements are forged integral with the bucket, these elements interlocking and forming a continuous shroud ring. The blade lengths and wheel diameters are made gradually larger, toward the exhaust

end, as shown. The blades are unsymmetrical, the exit angle being sharper than the entrance angle, the purpose of which is to reduce the exit steam velocity. The blades are flared at top and bottom so as to make the exit height of the steam passage greater than the inlet. The nozzles in the high-pressure stages are drilled in the diaphragm, while in the low-pressure stages rectangular nozzles formed in the

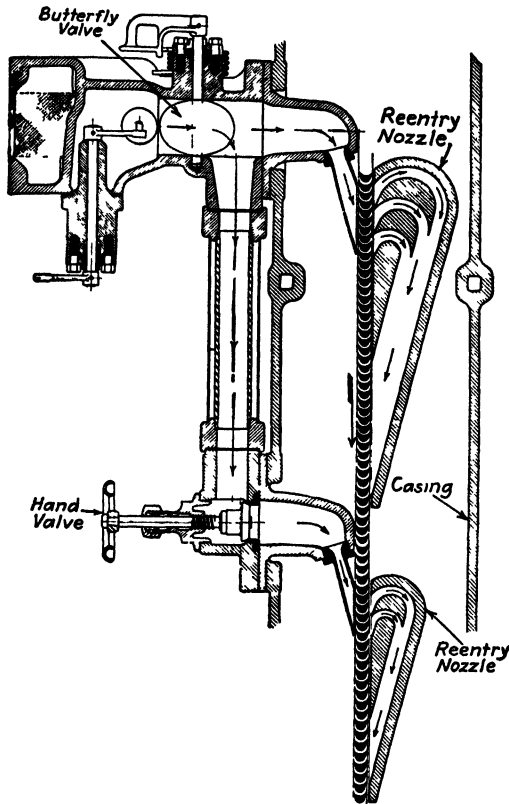



Fig. 223.—Diagrammatic view of a reentry impulse turbine. (*Dean-Hill Pump Company*)

diaphragms are used. Leakage along the shaft is prevented by carbon rings.

A Kingsbury thrust bearing, the principle of which is illustrated in Fig. 222, maintains the rotor of the Kerr turbine in proper position. This bearing is made up of a number of tilting and self-aligning bronze shoes, which bear against a collar on the turbine shaft. A film of oil is forced between the bearing surfaces.

232. Reentry Impulse Turbine.—This design of impulse turbine is illustrated in Fig. 223. The steam consumed by this turbine

enters the nozzle after passing through the governor-controlled inlet valve. In the nozzle, it expands to the final or exhaust pressure, and in doing so attains a maximum velocity. The kinetic energy of the steam, possessed by virtue of this velocity, is imparted, partially, to the rotor blades, as it makes the first pass through them. The steam then flows through the reversing or reentry chamber (Fig. 223) and is again directed onto the blades. After his second pass, the velocity of the steam is spent to a degree from which further attempt to extract energy would be worthless. It, therefore, then flows to the exhaust. The turbine illustrated in Fig. 223 is built in sizes ranging from 5 to 750 hp. and is very widely used.

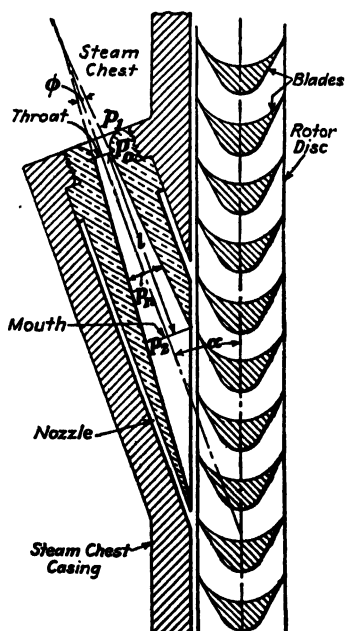



FIG. 224.—Diverging nozzle and blades for an impulse turbine.

Reentry turbines are confined to the so-called smaller sizes, and they are readily adaptable to driving a large variety of small equipment.

233. Impulse Turbine Nozzle.—The steam nozzle used in the construction of impulse turbines is the means through which the steam is permitted to expand and, thereby, transform heat energy into kinetic energy. If properly made and properly used, there will be a minimum of energy loss, and the issuing steam will be directed onto the turbine blades at an angle which will effect the greatest possible absorption of kinetic energy.

Steam, like gases, will expand instantaneously in the direction of least resistance when its pressure is suddenly reduced, as when it flows through an orifice. For this reason, it is often necessary to have the impulse nozzle relatively long, and with a diverging duct as shown in Fig. 224. The purpose of the divergence or taper is to effect a gradual expansion of the steam particles beyond the throat, with the aim of obtaining a flow from the mouth of the nozzle parallel to its axis.

234. Steam Velocity in Nozzles.—Under *ideal* conditions, there would be no energy losses from steam on flowing through a nozzle, and the energy transformation would be the result of a purely adiabatic change in the internal energy of the steam during the flow. The

final velocity of the steam, under these conditions, would be the maximum theoretically possible.

In developing an expression for the velocity of the steam, consider, first, an ideally perfect nozzle. By the law of conservation of energy, the total energy in a unit weight of steam at the nozzle exit is the same total energy it contained at the entrance. This may be summarized and expressed in equation form, as follows:

$$\frac{V_1^2}{2g} + 144p_1v_1 + u_1 + 778dq = \frac{V_2^2}{2g} + 144p_2v_2 + u_2 \qquad (136)$$

in which the symbols for the conditions of the steam at the nozzle entrance and exit, respectively, are as follows:

V_1 and V_2 = velocities, ft. per second.
p_1 and p_2 = pressures, lb. per square inch absolute.
v_1 and v_2 = specific volumes, cu. ft. per pound.
u_1 and u_2 = internal energy, ft.-lb. of work.
dq = heat change due to friction, etc., B.t.u. (= 0 for reversible adiabatic change).

In the actual nozzle, V_1 is extremely small compared with V_2, and, like dq, may be considered to be zero. The term $144pv + u$, in each case, equals $778h$, which changes the values from units of work to units of heat. Hence, Eq. (136) may be reduced to a more usable and general form, as follows:

$$\frac{V_2^2}{2g} = (h_1 - h_2)778$$

or

$$V_2 = 223.8\sqrt{h_1 - h_2} \qquad (137)$$

in which

V_2 = the final velocity of the steam, ft. per second.
h_1 = enthalpy of the steam at initial conditions, B.t.u. per pound.
h_2 = enthalpy of the steam after reversible adiabatic expansion, B.t.u. per pound.

On designing a nozzle, the friction between the steam and the nozzle should be considered. The effect of friction is a reheating of the steam at the expense of kinetic energy. Consequently, the final enthalpy h_2 actually is greater than in the case of the frictionless nozzle (ideal). It follows, also, that the final specific volume of the steam is greater, which necessitates a larger area at the mouth of the nozzle.

To correct the available energy $(h_1 - h_2)$ of Eq. (137) for friction loss, it is quite common to add a small percentage of the difference, based on experience, to h_2 when solving for the velocity. The amount of the correction may also be obtained by use of the following empirical equation, as given by various authorities:

$$q_R = 0.0006q_a(q_a - 45) \tag{138}$$

in which

q_R = reheat due to friction, B.t.u. per pound.

q_a = $(h_1 - h_2)$, available energy due to reversible adiabatic expansion [as in Eq. (137)], B.t.u. per pound.

The corrected value, h_2', for the final enthalpy of the steam becomes

$$h_2' = h_2 + q_R \tag{139}$$

in which h_2 and q_R are as before.

235. Critical Pressure in Nozzles.—The portion of the diverging nozzle (Fig. 224) between the steam chest and the throat is essentially an orifice and is subject to the same consideration as given to the flow of a perfect gas through an orifice (discussed in Art. 16, Chap. II). From the discussion in Art. 16, it may be stated that while the capacity of an orifice for discharging gaseous mediums depends, chiefly, on the area, the initial pressure of the medium, and the final pressure into which it is discharging, the *critical pressure* must also be taken into account.

The critical pressures for saturated and superheated steam on discharging through an orifice may be obtained by substituting the approximate values (1.14 and 1.28) for k in Eq. (33), page 28, and solving for p_o, in terms of p_1. Thus p_o, the critical pressure, may be expressed as follows:

$$\begin{aligned} p_o &= 0.58p_1 \text{ (for saturated steam)} \\ &= 0.55p_1 \text{ (for superheated steam)} \end{aligned}$$

When determining the steam velocity at the throat of a nozzle by use of Eq. (137), the final enthalpy of the steam h_2 must correspond to the critical pressure; that is, if the final discharge pressure is less than the critical value.

The steam velocity corresponding to the critical pressure is approximately 1,500 ft. per second and is sometimes called the *critical velocity*.

236. Nozzle Design.—The relation between the weight of steam discharged per unit of time and the cross-sectional area of a nozzle is expressed by Eq. (30), page 27, which, similarly, is as follows:

$$w_n v_n = A_n V_n \tag{140}$$

in which

w_n = weight of steam discharged, lb. per second.

v_n = specific volume of steam at point of consideration, cu. ft. per pound.

A_n = cross-section area, sq. ft.

V_n = velocity of the steam, ft. per second.

In Eq. (140), n signifies the point, in the nozzle, corresponding to the area under consideration. It should be noted that the velocity V_n is dependent on the pressure at the point n.

If the final discharge pressure p_2 is less than the critical pressure, in the design of a nozzle, the smallest cross-section area is at the throat, and the area at all points following diverge at a definite angle toward the mouth, as shown in Fig. 224. The pressure in the throat is the critical pressure, and the weight of steam to be discharged depends on the velocity, specific volume, and the area at this point. The area at any succeeding point n, toward the mouth of the nozzle, may then be determined on knowing the pressure p_n. However, the nozzle duct is usually made the frustum of a cone, and it follows, then, only to further determine the area at the mouth and the length l. The angle of divergence ϕ is determined from experience, and may be between 6 and 16 deg., depending on the conditions under which the nozzle is to be used. Knowing, ϕ, the length may be determined by means of the following right-triangle equation:

$$\tan \frac{\phi}{2} = \frac{d_2 - d_o}{2l} \tag{141}$$

in which

ϕ = the angle of divergence.

d_o = diameter at the throat, in.

d_2 = diameter at the mouth, in.

l = length from throat to mouth, in.

If ϕ is to be approximately 10 deg., the following equation may be used in solving for the approximate length, in inches:

$$l = \sqrt{15a_o} \tag{142}$$

in which

a_o = the area at the throat of the nozzle, sq. in.

If the area at the mouth of a diverging nozzle is made too small, there will result a further expansion of the steam after leaving, and a consequent loss of energy due to excessive friction in the nozzle and turbulence in the issuing jet. On the other hand, if the area at the mouth is too large, there is a similar loss due to excessive turbulence within the nozzle.

When the final discharge pressure is to be greater than the critical pressure, the nozzle may be purely cylindrical, or even converging toward the mouth.

The cross-section shape of a nozzle has little effect on its efficiency. Both circular and square shapes are used, depending on the service conditions of the nozzle.

The following example illustrates the use of the foregoing equations in determining the dimensions of a diverging nozzle to discharge a definite quantity of steam.

Example 12-1.—It is required to determine the diameter of the throat and mouth and the length of a nozzle for a single-stage impulse turbine, making use of the following data: steam discharged per hour, 3,000 lb.; initial and exhaust pressures, 200 and 16 lb. per square inch absolute; quality of steam to turbine, 0.98; angle of nozzle divergence (ϕ), 8 deg.

Solution.—The following conditions of pressure, etc., are obtained from the data given in the steam tables or on the Mollier diagram:

$p_1 = 200$	$p_o = 0.58 \times 200 = 116$	$p_2 = 16$
$x_1 = 0.98$	$x_o = 0.94$	$x_2 = 0.843$
$s_1 = 1.526$	$s_o = 1.526$	$s_2 = 1.526$
$h_1 = 1{,}181.33$	$h_o = 1{,}136$	$h_2 = 1{,}000$

The above values of x and h, for both critical and exhaust pressures, are for constant-entropy (adiabatic) expansion. From Eq. (138), the reheat during the pressure drop before the nozzle throat is

$$\begin{aligned} q_R &= 0.0006(1{,}181.33 - 1{,}136)[(1{,}181.33 - 1{,}136) - 45] \\ &= 0.009 \text{ B.t.u.} \end{aligned}$$

This value is so small that it may be neglected.
By Eq. (137), the velocity at the throat is

$$\begin{aligned} V_o &= 223.8\sqrt{1{,}181.33 - 1{,}136} \\ &= 1{,}507 \text{ ft. per second} \end{aligned}$$

The specific volume at the throat

$$v_o = 0.94 \times 3.846 = 3.615 \text{ cu. ft. per pound}$$

From Eq. (140), the area at the throat is

$$\begin{aligned} A_o &= \frac{3{,}000 \times 3.615}{3{,}600 \times 1{,}507} \\ &= 0.002 \text{ sq. ft. or } 0.288 \text{ sq. in.} \end{aligned}$$

Diameter at the throat = 0.605 in.

The reheat during the entire pressure drop is

$$\begin{aligned} q_R &= 0.0006(1{,}181.33 - 1{,}000)[(1{,}181.33 - 1{,}000) - 45] \\ &= 15 \text{ B.t.u.} \end{aligned}$$

The corrected value for the final enthalpy is

$$h_2' = 1{,}000 + 15 = 1{,}015 \text{ B.t.u. per pound}$$

From the Mollier diagram $x_2' = 0.859$. The final velocity

$$V_2 = 223.8\sqrt{1{,}181.33 - 1{,}015}$$
$$= 2{,}884 \text{ ft. per second}$$

Specific volume

$$v_2 = 0.859 \times 24.76 = 21.27 \text{ cu. ft. per pound}$$

The area at the mouth of the nozzle is

$$A_2 = \frac{3{,}000 \times 21.27}{3{,}600 \times 2{,}884}$$
$$= 0.006145 \text{ sq. ft., or } 0.885 \text{ sq. in.}$$

Diameter of the mouth = 1.06 in.

The length l is calculated by Eq. (141)

$$\tan 4° = \frac{1.06 - 0.605}{2l}$$
$$l = \frac{0.455}{2 \times 0.07}$$
$$= 3.25 \text{ in.}$$

237. Velocity Calculations for Single-stage Impulse Turbine.—Theoretically, the best results would be obtained if the nozzle of an impulse turbine were so placed that it discharged the steam in the same direction as the travel of the blades, and if the steam left the blades in the opposite direction to this. The blade velocity would then be one-half the steam velocity, and it would be so that all of the kinetic energy of the steam jet would be absorbed by the blades. The efficiency of the blades, under these conditions, would be 100 per cent.

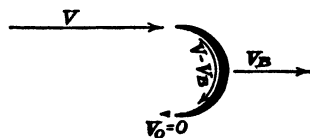



Fig. 225.—Theoretical impulse blade.

In Fig. 225, $V = 2V_B$. The inner face of the blade is a semicircle, and the velocity of the steam relative to the blade is $V - V_B$. Therefore, the velocity of the steam in the blade is also $V - V_B$, and the blade turns it in the opposite direction to that of entrance. The velocity of the steam leaving, relative to the blade, is $V - V_B$. To get the absolute velocity of the steam on leaving, another blade velocity must be subtracted; thus, $V_o = V - 2V_B = 0$. In this case, all of the steam velocity is absorbed by the blade: one-half on entrance, and one-half on leaving; and the blade efficiency is therefore, 100 per cent.

In the actual turbine, the position of the nozzle, as just discussed, would be impossible, as the nozzle would be in the line of travel of the moving blades. In practice, the axis of the nozzle is placed at an angle of approximately 20 deg. with the line of blade travel, as illustrated in Fig. 224, page 366.

A vector diagram can be used to aid in determining the blade velocity and the velocity at which the steam leaves the blades.

Figure 226 shows a theoretical vector diagram. The angle between the axis of the nozzle and the direction of blade travel is designated by α (approximately 20 deg.), and the lines AB and BC represent the components of the steam velocity V, perpendicular and parallel, respectively, to the direction of blade travel. The blade velocity V_B is equal to ½ of the parallel component. The relative velocity of the steam entering the blade is the vector resultant, V_1, of V and V_B. With no blade losses, V_2, the relative velocity of the steam leaving the blade, equals V_1 and the angles β_1 and β_2 are equal. The absolute velocity of the steam leaving the blade, V_o, is then the resultant of V_2 and V_B.

FIG. 226.—Velocity diagram, simple impulse turbine.

The efficiency of the blades in absorbing the kinetic energy of the steam, and the steam and blade velocities are expressed by the following equations:

$$\text{Efficiency} = \frac{\dfrac{wV^2}{2g} - \dfrac{wV_o^2}{2g}}{\dfrac{wV^2}{2g}}$$

$$= \frac{V^2 - V_o^2}{V} = 1 - (\sin \alpha)^2 \qquad (143)$$

The absolute velocity of the steam leaving the blade is

$$V_s = V \sin \alpha \qquad (144)$$

The blade velocity is

$$V_B = \frac{V \cos \alpha}{2} \qquad (145)$$

Theoretical horsepower imparted to blades is

$$\text{hp.} = \frac{w(V^2 - V_o^2)}{2g \times 550} \qquad (146)$$

In the above equations

w = weight of steam flowing, lb. per second.

g = the acceleration of gravity, ft. per second; and V, V_o and V_B are velocities, feet per second, as given in Fig. 226.

Example 12-2.—The turbine of Example 12-1 has the nozzle set at 20 deg. with the direction of blade travel; the r.p.m. is 20,000. Determine (a) the blade

efficiency, (*b*) the diameter of the rotor disc, and (*c*) horsepower to blades per pound of steam.

Solution.—Using the Eqs. (143) and calculating to (146), inclusive,

(*a*) $V_o = V \sin 20 \text{ deg.} = 2{,}884 \times 0.343 = 989$ ft. per second

$$\text{Blade efficiency} = 1 - (0.343)^2 = 0.883 \text{ or } 88.3 \text{ per cent}$$

(*b*) $$V_B = \frac{2{,}884 \times 0.94}{2} = 1{,}358 \text{ ft. per second}$$

Also

$$V_B = \frac{\pi DN}{60}$$

and

$$D = \frac{1{,}358 \times 60}{\pi \times 20\,000} = 1{,}295 \text{ ft., or } 15.54 \text{ in.}$$

(*c*) $$\text{hp.} = \frac{.833(2{,}884^2 - 989^2)}{64.4 \times 550} = 173$$

238. Velocity Calculations for Multiple-velocity-stage Turbines.—The nozzle of the velocity-stage impulse turbine is of the same design as for the single-stage turbine; and, similarly, all the pressure drop occurs in one set of nozzles.

The blade velocity, in this case, may be taken as

$$V_B = \frac{V \cos \alpha}{2n} \quad \text{[similar to Eq. (145)]} \quad (147)$$

in which

V_B and V = velocities as indicated in Fig. 227, ft. per second.

α = angle between the nozzle axis and the direction of blade travel.

n = number of velocity stages.

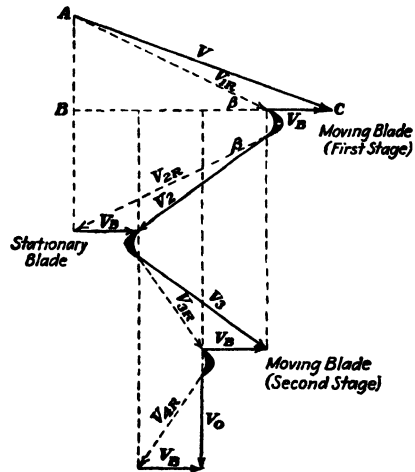



Fig. 227.—Velocity diagram for an impulse turbine (two velocity stages).

The vector diagram may be constructed in a manner similar to that shown in Fig. 226, which is the vector diagram for a single-stage turbine. Figure 227 is the velocity diagram for a turbine of two velocity stages. The nozzle angle is 20 deg.

The blade velocity is ¼ of *BC*, and the resultant of V and V_B is V_{1R}, which is the relative velocity of the steam entering the blades of the first stage. Assuming zero friction in the blades V_{2R}, the rela-

tive steam velocity of exit is equal to V_{1R} and makes the same angle β with the blade travel. V_2, then, becomes the absolute velocity of exit. The same procedure for the second stage gives V_{3R} and V_{4R} as the relative velocities of entrance and exit, respectively, and V_o as the absolute velocity of exit from the last stage. All of the components of the steam velocity, in the direction of the blade motion, have been imparted to the blade as the steam leaves the last stage. Thus V_o, the absolute velocity of exit is equal to AB and is equal to $V \sin \alpha$, as in Eq. (144), and the blade efficiency is as given in Eq. (143).

Example 12-3.—Considering the turbine of Example 12-1 to be an impulse turbine with three velocity stages; nozzle angle 20 deg.; diameter of rotor 36 in., determine (*a*) nozzle dimensions, (*b*) blade efficiency, and (*c*) r.p.m. of rotor, neglecting friction.

Solution.—*a*. Since the nozzle is the same as that for the simple impulse turbine, the size is as given in Example 12-1. Thus

$$d_o = 0\ 605 \text{ in. (throat)}$$
$$d_2 = 1.06 \text{ in. (mouth)}$$
$$l = 3.25 \text{ in.}$$

b. Solving for blade efficiency:

$$V = 2{,}884 \text{ ft. per second (Example 12-1)}$$
$$V_o = V \sin 20 \text{ deg.} = 2{,}884 \times 0.343 = 989 \text{ ft. per second}$$
$$\text{Efficiency} = 88.3 \text{ per cent (Example 12-2)}$$
$$V_B = \frac{2{,}884 \times 0.94}{2 \times 3} = 452.6 \text{ ft. per second}$$

c.
$$N = \frac{452.6 \times 60}{\pi \times 3} = 2{,}880 \text{ r.p.m.}$$

239. Exemplary Calculations for Impulse Turbines.—The following examples illustrate practical methods used in an analysis of multistaged, impulse turbines. There are certain assumptions in various places, which turbine designers have access to, and which have been compiled through years of designing, manufacturing and testing.

In the calculations of the steam expansion through the turbine, when the distribution of the theoretical available energy and the efficiencies of the successive stages are given, the following general method is used. First determine the pressure at each stage by the constant entropy expansion that would occur in the ideal turbine. Then these pressures will remain unchanged in the real turbine. The per cent reheat for each stage will be obtained by subtracting the stage efficiency from 100 per cent. Each stage will be worked in succession starting with the high-pressure stage. The available energy for each stage will be obtained by the constant-entropy expansion from the pressure at the entrance to that at the exit. By adding the reheat to the enthalpy at the exit, at constant pressure, the enthalpy and

entropy of the steam leaving that stage are obtained. These are also the conditions of the steam entering the next stage. The following examples illustrate these calculations.

Example 12-4.—A six-stage, impulse turbine is supplied with steam at a pressure of 200 lb. per square inch gage; 150° of superheat; condenser vacuum 28.21 in. of mercury; barometer 29.64 in. of mercury. The net load carried by the turbine shaft is 20,000 kw.; shaft speed 1,500 r.p.m. The mean blade speed is 420 ft. per second and the energy distribution required is 30 per cent in the first stage and 14 per cent in each of the following stages. The efficiency of the first stage is assumed to be 48 per cent (that is, 48 per cent of the available heat energy is converted to kinetic energy), and that for each of the succeeding stages 55 per cent. Determine the items in the following:

Calculations.—*a*. Steam rate per horsepower-hour of the perfect turbine:

$p_1 = 200 + 14.55 = 214.55$ $\quad p_2 = 1.43 \times 0.491 = 0.7$ lb. per square inch absolute.
$h_1 = 1{,}285$ B.t.u.

At constant entropy $h_1 - h_2 = 1{,}285 - 893 = 392$ B.t.u. (2,545 B.t.u. per horsepower-hour) ÷ 392 B.t.u. per pound = 6.5 lb. of steam per horsepower-hour.
(NOTE.—The steam rate by test was 11 lb. per kilowatt-hour, or 8.2 lb. per horsepower-hour.)

b. Efficiency ratio based on the Rankine cycle:
6.5 ÷ 8.2 = 0.793 or 79.3 per cent, using the actual steam rate given in *a*.

c. Mean blade diameter (rotor):

$$3.1416 \times 1{,}500 \times D = 420 \times 60$$
$$D = 5.35 \text{ ft. or } 64.2 \text{ in.}$$

d. Constant-entropy expansion in each stage (ideal turbine):
Enthalpy change = 392 B.t.u. per pound.
30 per cent of 392 = 117 B.t.u. per pound, first stage.
14 per cent of 392 = 55 B.t.u. per pound, each remaining stage.

e. Actual steam path through the turbine:
(1) To determine pressure of steam leaving each stage.

Stage	Constant-entropy expansion	
	Enthalpy leaving, B.t.u. per lb.	Steam pressure leaving, lb. per sq. in.
1	1,168	60
2	1,113	29
3	1,058	13
4	1,003	5.5
5	948	2
6	893	0.7

(2) To determine the enthalpy of steam leaving each stage, by calculating the constant-entropy expansion and reheat for each stage.

Stage	B.t.u. per lb.		
	Enthalpy change at constant-entropy	Reheat	Enthalpy leaving
1	117	61	1,229
2	61	27.5	1,195.5
3	61.5	27.5	1,161.5
4	59.5	27	1,129
5	65	29	1,093
6	63	28.5	1,058.5

From the above calculations, the enthalpy change through the turbine is 1,285 − 1,058.5 = 226.5 B.t.u.

f. Efficiency ratio based on the above values:

$$226.5 \div 392 = 0.578 \text{ or } 57.8 \text{ per cent}$$

g. Water rate based on the above values:

$$2{,}545 \div 226.5 = 11.22 \text{ lb. of steam per horsepower-hour}$$

h. Weight of steam used:

$$(20{,}000 \times 11.22) \div (0.746 \times 3{,}600) = 83.5 \text{ lb. per second}$$

Summary.

Stage	1	2	3	4	5	6
Enthalpy leaving B.t.u. per pound	1,229	1,195.5	1,161.5	1,129	1,093	1,058.5
Blade speed, ft. per second	420	420	420	420	420	420
Stage efficiency, per cent	48	55	55	55	55	55
Available energy, B.t.u. per pound	117	61	61.5	59.5	65	63
Enthalpy after constant-entropy expansion, B.t.u. per pound	1,168	1,168	1,134	1,102	1,064	1,030
Reheat, B.t.u. per pound	61	27.5	27.5	27	29	28.5
Pressure at exit, lb. per square inch absolute	60	29	13	5.5	2	0.7
Temperature, °F	390	310	235			
Quality at exit, per cent	......	......	......	99.7	97.8	96.0
Superheat at exit, °F	97	62	29			
Pressure at entrance, lb. per square inch absolute	214.55	60	29	13	5.5	2
Superheat at entrance, °F	150	97	62	29		
Saturation temperature, °F	387.8	292.7	248.4	206	166.3	126.1
Steam temperature, entrance, °F	537.8	390	310	235		
Throat pressure in nozzle lb. per square inch absolute	118	33	16.2	7.2	3.1	11.4
Heat drop to throat, B.t.u. per pound	57	51	48	45	38	34

Example 12-5.—A test conducted on a vertical turbine with five pressure stages gave the following results:

a. Load carried, 10,087 kw.
b. Steam pressure at throttle, 205.4 lb. per square inch absolute.
c. Superheat at throttle, 128.7°F.
d. Barometer, 29.34 in. of mercury.
e. Condenser vacuum, 27.8 in. of mercury.
f. Dry steam 14.14 lb. per kilowatt-hour.
g. Turbine speed, 720.7 r.p.m.
h. Rotor diameter at mean height of blades, 11⅔ ft.
i. Steam condition leaving each stage:

Stage	1	2	3	4	5
Pressure, lb. per square inch absolute	63.6	25.4	8.1	2.79	.75
Superheat, °F	67.1	0			
Quality, per cent	....	100	98.2	96.3	93.4

From the above data, determine the following:

Calculations.—*a.* Change in enthalpy per pound of steam, through the turbine.

Initial enthalpy of steam	1,272 B.t.u.
Enthalpy at exit of first-stage nozzles (adiabatic)	1,167
Enthalpy change at constant entropy	105
Reheat in first stage (1,212 − 1,167)	45
Enthalpy to second-stage nozzles	1,212
Enthalpy	1,141
Enthalpy change at constant entropy, second stage	71
Reheat, second stage (1,161 − 1,141)	20
Enthalpy to third-stage nozzles	1,161
Enthalpy after adiabatic drop	1,080
Enthalpy change at constant entropy, third stage	81
Reheat, third stage (1,122 − 1,080)	42
Enthalpy to fourth-stage nozzles	1,122
Enthalpy	1,053
Enthalpy change at constant entropy, fourth stage	69
Reheat, fourth stage (1,084 − 1,053)	31
Enthalpy to fifth-stage nozzles	1,084
Enthalpy	1,007
Enthalpy change at constant entropy, fifth stage	77
Reheat, fifth stage (1,032 − 1,007)	25
Enthalpy per pound of exhaust steam	1,032 B.t.u.

Actual change in enthalpy per pound of steam in passing through the turbine:

$$1{,}272 - 1{,}032 = 240 \text{ B.t.u.}$$

b. Enthalpy of exhaust steam, assuming constant-entropy expansion, from initial to final pressure:

893 B.t.u.

c. Energy available, assuming constant-entropy expansion:

$$1{,}272 - 893 = 379 \text{ B.t.u.}$$

d. Efficiency ratio of turbine based on the above results:

$$240 \div 379 = 0.633 \text{ or } 63.3 \text{ per cent}$$

e. Dry steam consumed per hour:

$$10{,}087 \times 14.14 = 144{,}000 \text{ lb.}$$

f. Horsepower developed:

$$10{,}087 \div 0.746 = 13{,}500$$

g. Dry steam consumed:

$$144{,}000 \div 13{,}500 = 10.66 \text{ lb. per horsepower-hour}$$

h. Dry steam consumed by the perfect turbine, working on the Rankine cycle:

$$2{,}545 \div 379 = 6.72 \text{ lb. per horsepower-hour}$$

i. Efficiency ratio of the turbine, based on the steam rate:

$$6.72 \div 10.66 = 0.631 \text{ or } 63.1 \text{ per cent}$$

j. Efficiency ratio (the ratio of the heat equivalent of the work done to the heat available from the Rankine cycle, both as B.t.u. per pound of steam):

$$(10{,}087 \times 3{,}414) \div (379 \times 10{,}087 \times 14.14) = 0.637 \text{ or } 63.7 \text{ per cent}$$

Summary.

Stage	1	2	3	4	5
Enthalpy at nozzle entrance, B.t.u. per pound	1,272	1,212	1,161	1,122	1,084
Mean blade speed, ft. per second	445	445	445	445	445
Stage efficiency, per cent	57.2	71.4	49.2	55	67.5
Available energy, B.t.u. per pound	105	71	81	69	77
Enthalpy after constant-entropy drop, B.t.u. per pound	1,167	1,141	1,080	1,053	1,007
Reheat, per cent	42.8	28.6	51.8	45	33
Pressure leaving nozzles, lb. per square inch absolute	63.6	25.4	8.1	2.8	.8
Temperature leaving nozzles, °F	363.7	240.9	183.5	153.5	92
Quality, leaving nozzles, per cent	....	100	98.2	96.3	93.4
Superheat, leaving nozzles, °F	67.1				
Steam velocity, leaving nozzles, ft. per second	2,850	1,844	1,960	1,790	1,882

Figure 228 shows a diagram of the pressure and velocities of the steam through the various stages of the turbine of Example 12-5.

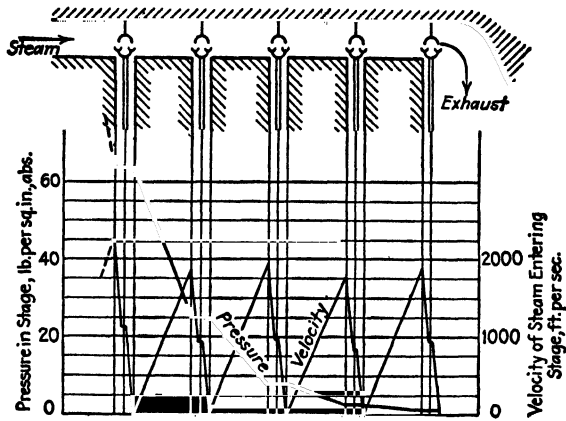



FIG. 228.—Pressure-velocity diagram.

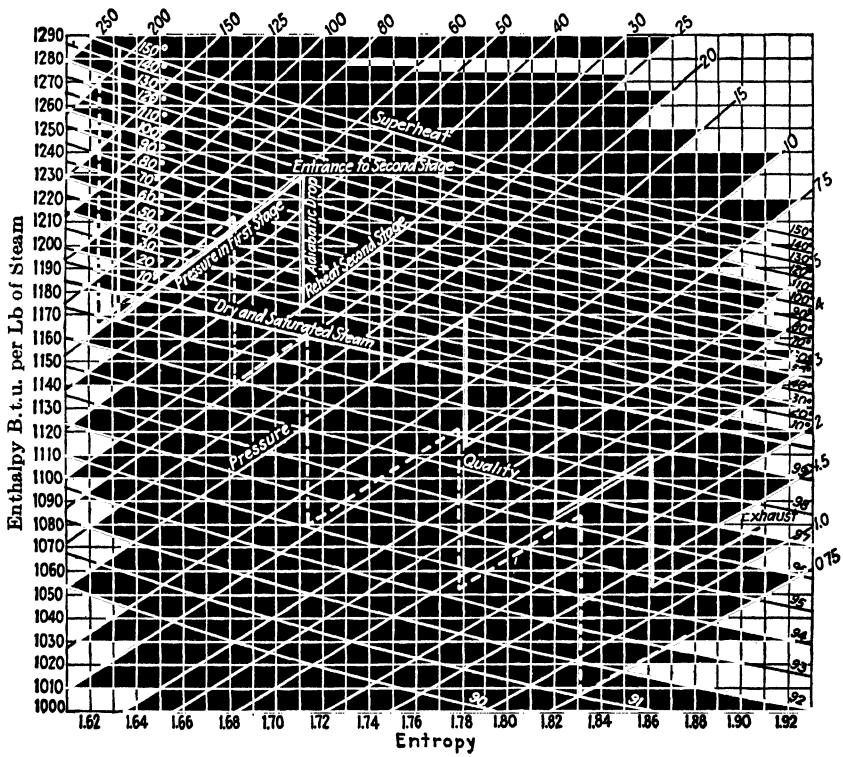



FIG. 229.—Steam expansion plotted on Mollier diagram.

It can be noted that the pressure drop is not equal for each of the five stages, but is such that will give approximately the same velocity of the respective steam jets. This, of course, is for a turbine having the same mean blade diameter for all stages. In Fig. 229, both problems are located on the Mollier diagram. This shows the result of friction in the nozzle, and blades, and the variation of the steam expansion from a constant entropy path.

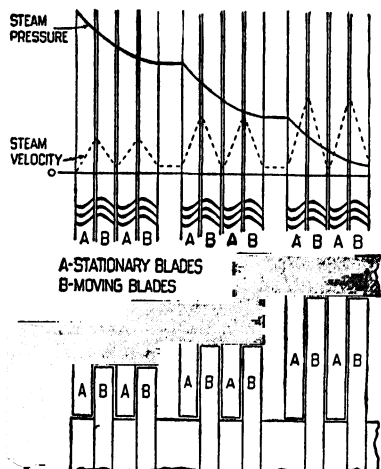



FIG. 230.—Pressure and velocity diagram, reaction turbine.

240. Reaction Turbines.—Figure 230 illustrates, diagrammatically, the expansion of steam through the blades of a reaction turbine. Alternate rows of moving blades are attached to the rotating drum, and the rows of stationary blades are attached to the casing. The blades are shaped so that their effect on the steam is that of a nozzle. The steam, in each row of moving blades, is divided into a large number of small steam jets. Each jet tends to acquire a high velocity, resulting in a backward or reactive force on the moving blades. Likewise, these jets of steam on leaving the stationary blades have an increased velocity due to expansion, and impinge on the following row of moving blades. As the pressure drop in each row of blades is small, many rows of both stationary and moving blades are necessary to properly attain the total pressure drop between the initial and exhaust pressures. Reaction turbines are thus designed to utilize both the impulse and reactive forces of the steam jets.

The principal builders of axial-flow, reaction turbines in this country are the Allis-Chalmers Manufacturing Company, the Westinghouse Manufacturing and Electric Company, and the American Brown-Boveri Company.

The Allis-Chalmers turbine, for generator drive, is of the horizontal reaction type as shown in Fig. 231. The rotor consists of a hollow drum built up in three steps, *H*, *J* and *K*. Steam enters through pipe *C*, after passing through governor valve (not shown), and flows through a passage entering the blading at *E* around the entire circumference of blading. It then flows parallel to the axis of the turbine through the alternate rows of moving and stationary blades, leaving the blading at *F*, through the exhaust pipe, and flowing to the condenser.

To provide for the increased volume of the steam, as it expands, the spindle is divided into three different steps, and on each step, the blades are arranged in groups of increasing lengths.

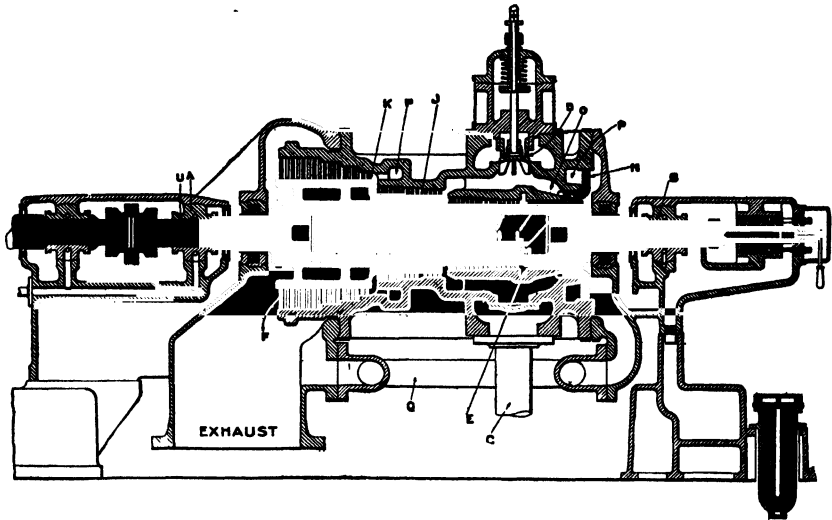



Fig. 231.—Sectional view of Allis-Chalmers reaction turbine.

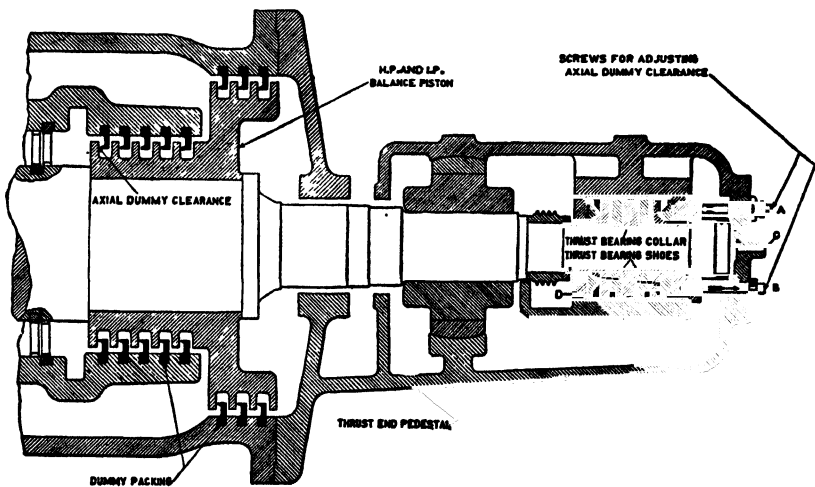



Fig. 232.—Showing dummy pistons and adjustable bearing, Allis-Chalmers turbine.

The thrust of the steam at each spindle step is counteracted by the balance pistons *L*, *M* and *N*, which are of the correct diameter to neutralize the thrust on the steps *H*, *J* and *K*, respectively. In order that the proper pressure may act on the balance pistons, equalizing

passages *O*, *P* and *Q* are provided, connecting the balance pistons with the proper stage of blading.

Leakage of steam around the outside of the balance pistons is retarded by the use of labyrinth packings (Fig. 232). In this construction the piston and casing are provided with alternate rings,

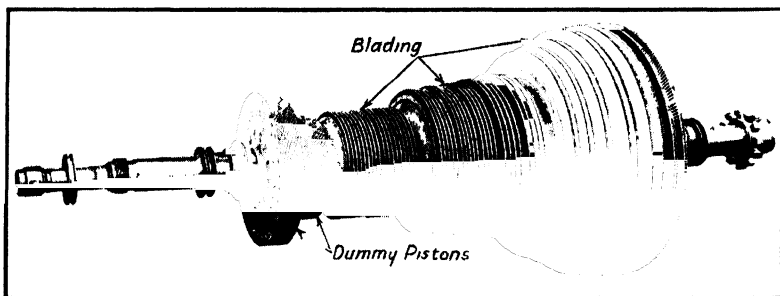



Fig. 233.—Rotor of Allis-Chalmers reaction turbine.

and the successive throttling effects hinder the flow of steam past the balance pistons.

The position of the spindle is regulated by a small adjustable collar bearing inside the housing of the main bearing. This adjustable bearing is used to locate and hold the spindle in position in relation to the casing for the proper clearances between the rings of the labyrinth packing.

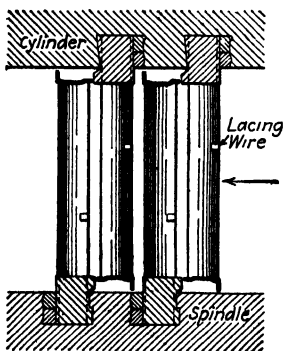



Fig. 234.—Showing blade construction, Allis-Chalmers reaction turbine.

Where the shaft leaves the casing, leakage is prevented by the use of water-sealed glands. The gland consists of a paddle wheel fixed on the spindle and rotating in a special V-shaped casing. Water is introduced in this casing, and, owing to the high speed, the paddle fills the groove around the rim of the wheel, forming a perfect seal.

Valve *D* (Fig. 231) is the overload valve, admitting high-pressure steam to the intermediate stage on step *J*.

Figure 233 shows a rotor assembled, with complete blading, showing clearly the three different diameters of the spindle. The rings of the labyrinth packing on the balance pistons are clearly seen.

The Allis-Chalmers blading consists of drawn metal strips of uniform cross-section, made into a rigid segment by casting the roots of the blades into a segment of the foundation ring (Fig. 234). The top ends of the blades are brazed to a channel-shaped shroud ring, which

has the central grooved portion indented toward the blades to hold the ends firm, thus assuring the proper angle and spacing. The flanges of the shroud ring are made thin so that, in case of contact, there will be no dangerous heating, and no blades torn out should they happen to strike during operation. All blades 3 in. or longer have a lacing strip to aid in producing a structure which is rigid against vibration. The segment of blading is placed in a groove in the spindle which is undercut to fit a projection that is machined on the foundation ring. The segment is then locked into the groove by means of a calking strip, which is upset in the groove, thus holding the ring in place.

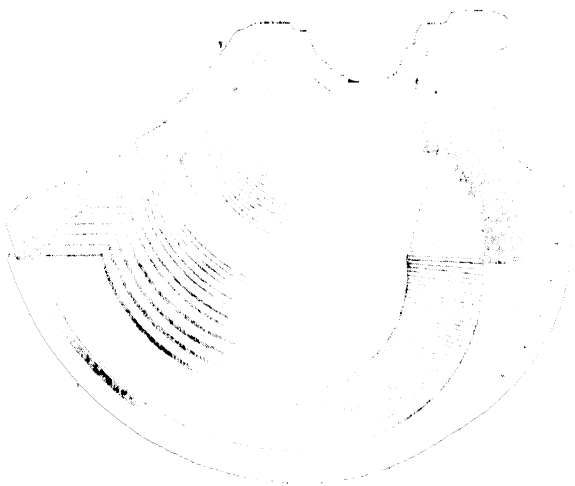



Fig. 235.—Top half of casing, Allis-Chalmers turbine

Figure 235 shows a photograph of the top half of the casing of the Allis-Chalmers reaction turbine. This clearly shows the arrangement of the stationary blading.

241. Combination Impulse and Reaction Turbines.—Turbines of this type are designed with the high-pressure stages of the impulse type and those following of the reaction type. The reaction stages receive the steam at a reduced pressure and absorb the energy resulting from further expansion to the exhaust pressure. Such an arrangement is of advantage when high-pressure and high-temperature steam is to be used. The reason for this is that impulse elements, under these conditions, eliminate steam leakage around the blade tips.

Figure 236 shows a sectional view of a 60,000-kw., Westinghouse turbine of the combined impulse and reaction type. It is designed to take the exhaust steam from two other turbines which receive

steam at 1,300-lb. pressure. The exhaust pressure is 290 lb., and, before the steam is delivered to the low-pressure turbine, it is resuperheated to 700°F. Extraction of steam at three points, for feedwater heating, is provided for in this turbine.

The rotor of the turbine consists of two forgings: one of carbon steel, at the high-pressure end, and one at the exhaust end of 3.5 per cent nickel-steel. Nickel-steel is used because of the high blade load at this end. There are 2 velocity stages of impulse blading for the first 2 stages of pressure drop, and, following these, are 22 stages of reaction blading. The last three rows of reaction blading are carried on integral

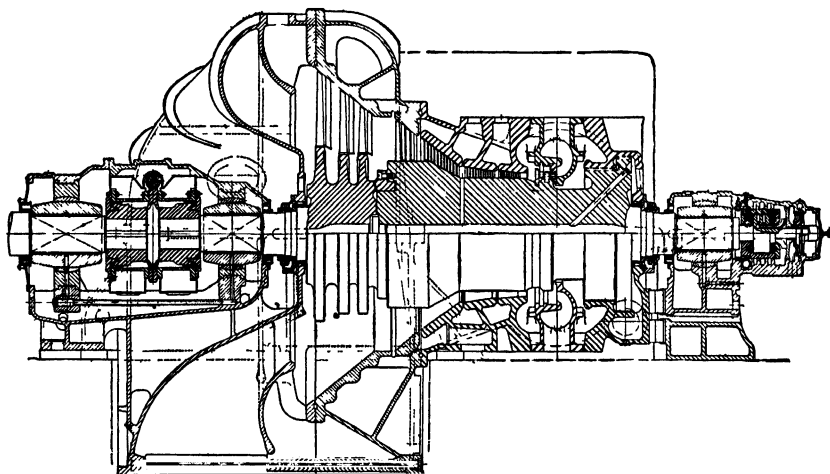



Fig. 236.—Westinghouse turbine, 60,000-kw. capacity.

discs. The blade grooves consist of a series of curved slots which are cut in an axial direction. Each blade, inserted, locks the one ahead of it, and projections on the ends of the blade roots are afterward rolled-over into grooves in the side of the discs.

To prevent distortion during cooling, a turning gear, capable of rotating the spindle at a speed of about 24 r.p.m., is provided. In addition, compressed air connections are attached to the cylinder to circulate the vapors therein during the cooling period.

The data in Table 12-3 give an idea of the size of this unit.

242. Special Industrial Turbines.—Under *industrial turbines* may be listed the following:

1. Bleeder turbine.
2. Low-pressure turbine.
3. Mixed-pressure turbine.
4. Back-pressure turbine.

The function of such turbines, aside from power generation, is to supply steam for manufacturing processes or heating or to utilize steam that would otherwise be wasted.

243. Bleeder Turbines.—A distinction may be made between *bleeder turbines* and *extraction turbines.* An extraction turbine is of the type used in central stations, in which steam is extracted at differ-

TABLE 12-3.—PHYSICAL DATA ON UNIT AND AUXILIARIES FOR 60,000-KW WESTINGHOUSE TURBINE (FIG 236)

Item	Value
Rating at 0 85 power factor, kw	60,000
Steam pressure at throttle, lb gage	290
Steam temperature, °F	700
Speed, r p m	1,800
Impulse blading, stages	2
Reaction blading, stages	22
Generator, 3 phase, 60 cycle, 13,800 volt, kw	60,000
Exciter, direct connected, 250 volt, kw	175
Air-cooler surface, sq ft	15,080
Length of unit, overall	63 ft ½ in
Width of turbine	21 ft 10 in
Width of generator	13 ft 8 in
Height of turbine, from floor	12 ft 11 in
Height of generator from floor	11 ft 2⅝ in
Distance, center to center, turbine bearings	17 ft 8 in
Distance, center to center, generator bearings	25 ft 11 in
Diameter of turbine shaft, in	14
Diameter of generator shaft, in	16
Weight of turbine rotor, tons	59
Weight of generator rotor, tons	73
Weight of turbine generator, tons	466
Diameter of largest wheel	12 ft 2 in
Peripheral speed of largest wheel, ft per second	1,142
Condenser and Auxiliaries	
Maker	Foster-Wheeler Corp
Condenser, single pass, 9,240 tubes, ⅞ in diam, 26 ft long; area condensing surface, sq ft	55,000
Circulating pumps, two, g p m each	52,500
Drive, motor, hp	400
Condensate pumps, two, g p m each	1,400
Drive, motor, hp	100
Air pumps, c f m at 0 67 in absolute	15
Extraction heaters, four, sq ft No 1-2, 700; No 2-2, 500; No 3-2, 900; No 4-2, 650	
Extraction heater pump, one, 4 in, g p m	370

ent stages and used in heating the feedwater for the boiler, in successive steps. There is no attempt made to regulate the pressure in the extraction lines, and it is allowed to vary with changes in pressure of the steam leaving the throttle. Such an extraction turbine is shown in Fig. 236. A bleeder turbine usually has one bleeder line

which supplies steam for a manufacturing process. In this type, a governing system is provided for keeping the pressure in the bleeder line constant. The principal function of the bleeder turbine is to keep the pressure constant in the bleeder line despite all variations in the demand for steam, and, in addition, mechanical energy is obtained during the expansion of the steam in the turbine. The mechanical or electrical output may or may not be kept constant, depending upon the design of turbine. The bleeder turbine is usually a condensing turbine operating on high-pressure steam.

The Terry bleeder turbine is illustrated in Fig. 237. It consists essentially of two turbines on a common shaft. There is a non-

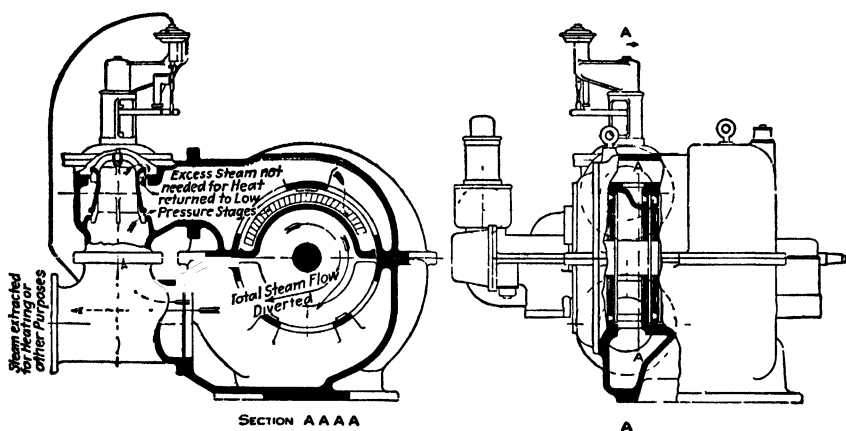



Fig. 237.—Terry bleeder mechanism

condensing element and a condensing element. The high-pressure steam passes through the non-condensing element, then strikes the solid disc of the bleeder diaphragm and is all diverted to the bleeder outlet. If the quantity of steam thus diverted is in excess of that required to maintain the pressure in the bleeder line, thereturn valve is opened and allows the excess steam to return to the turbine and pass through the low-pressure stages. This arrangement permits any quantity, up to 100 per cent of the steam flow, to be bled from the turbine at any desired pressure, depending upon the throttle pressure.

In the Fig. 237, a small pipe connects the bleeder line to a pressure-regulating valve in the return steam line. With an increase in bleeder-line pressure, the oil relay valve admits oil under pressure below the piston in the operating cylinder. This opens the bleeder valve, admitting steam to the low-pressure stages, thus reducing the bleeder-line pressure.

The bleeder valve is slightly unbalanced and acts as an automatic relief valve to the bleeder line. In case of failure of the control, an excess of pressure in the bleeder line will automatically open the bleeder valve, thereby preventing damage.

The bleeder mechanism used on General Electric turbines is slightly different and is illustrated in Fig. 238. The bleeder valve is of annular gridiron form and bears against a stationary diaphragm with openings, as shown, corresponding with the ports in the revolving

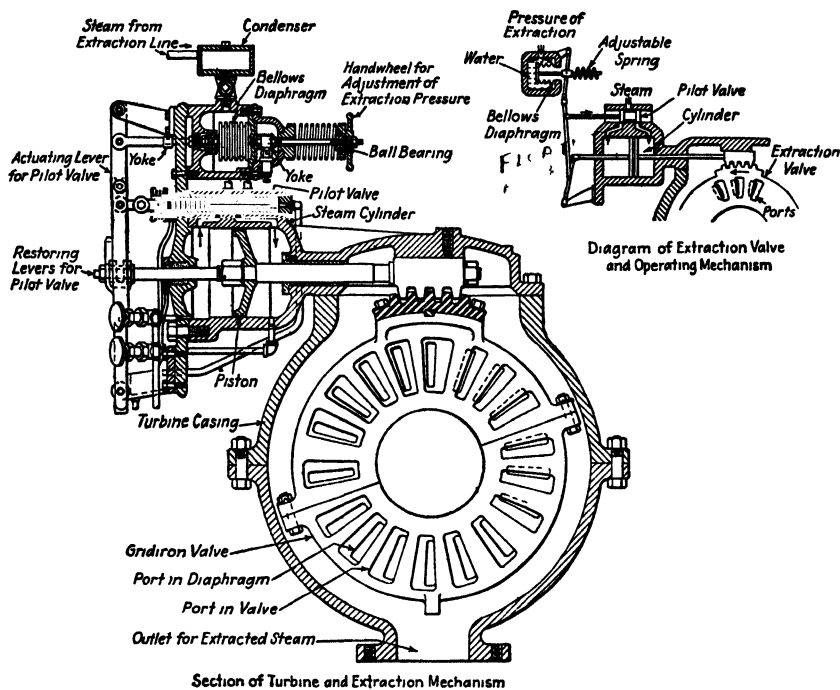



FIG. 238.—General Electric bleeder mechanism.

disc. The more the ports are open the more steam passes through the lower stages of the turbine, tending to reduce the pressure in the bleeder line. Closing the ports has the opposite effect.

A small pipe connects the bleeder line with the sheet-metal bellows in the diaphragm chamber of the pressure-regulating mechanism. A small condenser is provided to insure that the chamber surrounding the bellows is always filled with water. The bleeder-line pressure in this condenser acts on the outside of the bellows. The inside is open to the atmosphere. With an increase of pressure in the bleeder line, the bellows is compressed (Fig. 238) and, through a system of levers, this movement acts on the pilot valve which admits steam to

the operating cylinder. The piston of the operating cylinder is thus moved and rotates the gridiron valve to open the ports which allow more steam to flow to the low-pressure stages of the turbine. This, of course, reduces the flow to the bleeder line and restores the bleeder pressure to normal. After each change, the pilot valve is restored to a neutral position by the operating linkage.

The spring, as shown in the figure, opposes the thrust pressure of the bellows. By a handwheel adjustment, the spring force can be changed, thereby making desired changes in the bleeder-line pressure. The operating cylinder has a diameter of 8 in., so a steam pressure of

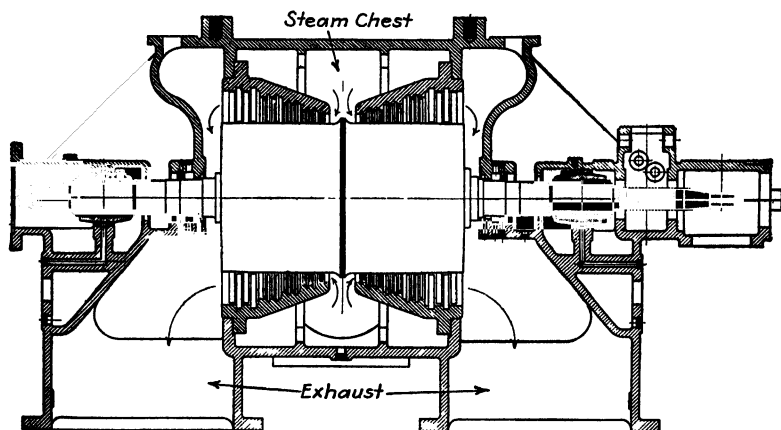



Fig. 239.—Westinghouse low-pressure reaction turbine.

100 lb. per square inch will produce a force of 5,000 lb. to move the gridiron valve. The ports in gridiron disc and stationary diaphragm are arranged in groups that open and close in sequence. This corresponds with the grouping of the nozzles of the first stage of the turbine.

244. Low-pressure Turbines.—Turbines of this class take steam at about atmospheric pressure and expand it to a high vacuum. These turbines are well adapted to handle large volumes of low-pressure steam and are often used in connection with the exhaust of a noncondensing, reciprocating engine. They are also used in shops where many engines are employed for operating equipment such as rolling mills, etc., as such equipment provides a large supply of low-pressure steam.

Figure 239 shows a sectional view of a typical Westinghouse low-pressure, reaction turbine. The characteristic feature of this turbine is the simplicity of its design. Low-pressure steam enters at the center and flows horizontally, in opposite directions, through the reaction blading, exhausting at each end of the casing.

If the supply of steam for low-pressure turbines is intermittent, such as that from rolling-mill engines, a steam accumulator can be used to good advantage.

The accumulator consists of a closed vessel for storing a large quantity of water under pressure. Used in connection with a low-pressure turbine, the excess steam from the supply flows into the vessel where it condenses and is absorbed by the water. This occurs under increasing pressure and keeps the water near the temperature of saturation. When the supply of steam ceases, the steam pressure decreases, and, as a result, a part of the hot water vaporizes. Thus, a continuous supply of steam is available for the low-pressure turbine used in connection with the accumulator. The pressure gradually

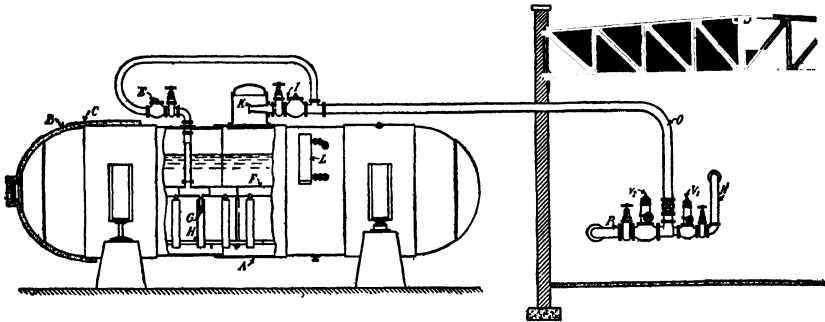



Fig. 240.—Ruths steam accumulator.

decreases until the engine or other equipment again begins to furnish steam. The excess steam is utilized to build up the accumulator temperature and pressure. There is usually a provision for supplying steam from the boiler through a reducing valve in case the pressure in the accumulator gets too low.

Figure 240 shows an out-of-door accumulator installation. The tank *A* is made of riveted steel plate and is about 90 per cent full of water. Around the tank is a covering of insulating material *B* protected by weatherproof covering *C*. The non-return valve *E* admits charging steam to the internal steam-distributing pipe *F* and the charging nozzles *G*. These are equipped with circulation pipes *H* to insure uniform and noiseless heating of the water.

The discharge of steam from the nozzle passes through a non-return valve *I*. A De Laval nozzle *K* limits the amount of the discharge in case of any emergency. A water column *L* shows the water level, and line *O* is the line through which the steam flows both to and from the accumulator. *N* indicates the high-pressure line from the boiler, and V_1 and V_2 are the automatic regulating valves. If the

accumulator is up to maximum pressure, further input of steam is stopped. Or if fully discharged, then boiler steam is admitted at reduced pressure to the accumulator through a secondary line *P*.

The accumulator is used in power plants of industries that require a variable supply of steam for their factory processes, particularly the pulp and paper industry, textile, chemical, rubber industry and others.

245. The Mixed-pressure Turbine.—The mixed-pressure turbine is used where there is a variable supply of low-pressure steam. This exhaust steam is sent through the low-pressure stages of a condensing turbine. The turbine governor automatically limits its consumption

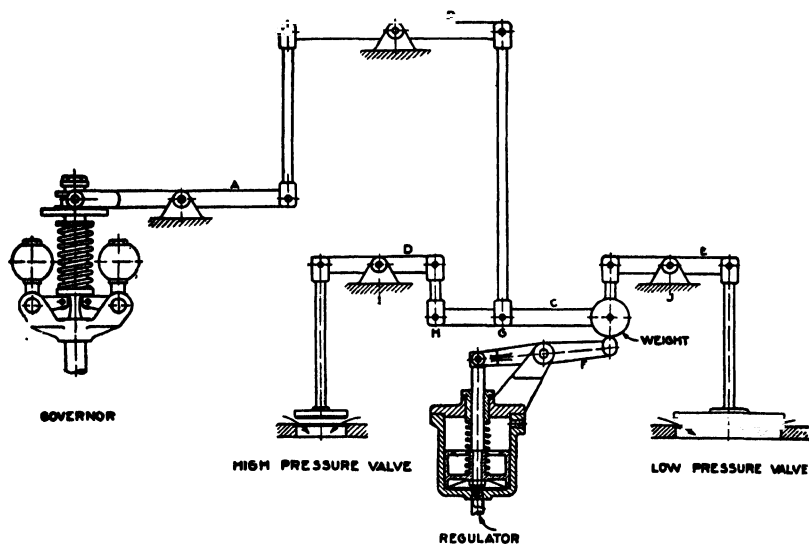



FIG. 241.—Diagrammatic arrangement of governor and valves of a mixed-pressure turbine.

of high-pressure steam and gives preference to the low-pressure steam. However, when sufficient low-pressure steam is not available, enough high-pressure steam is admitted to the first stages to carry the load. This steam is expanded through the high-pressure and low-pressure elements as in the ordinary condensing turbine.

The operation of the *Kerr governor for mixed-pressure turbine* is shown by diagram in Fig. 241. In the position shown, there is load on the mixed-pressure turbine, and both high-pressure and low-pressure valves are open. The cylinder under the regulator piston is connected by a pipe to the low-pressure steam line. If the low-pressure steam supply diminishes, due to the reduction of pressure, the regulator piston moves downward. This raises the weighted end of the lever

C, closing the low-pressure valve and opening the high-pressure valve to make up for the shortage of low-pressure steam. If the low-pressure steam supply increases, the reverse action takes place; but, if the low-pressure steam supply exceeds the requirements, when the high-pressure valve is closed, the turbine speeds up, and the governor weights, by centrifugal force, lift the lever *C*, which closes the low-pressure valve. With both valves open, a decrease in the load will

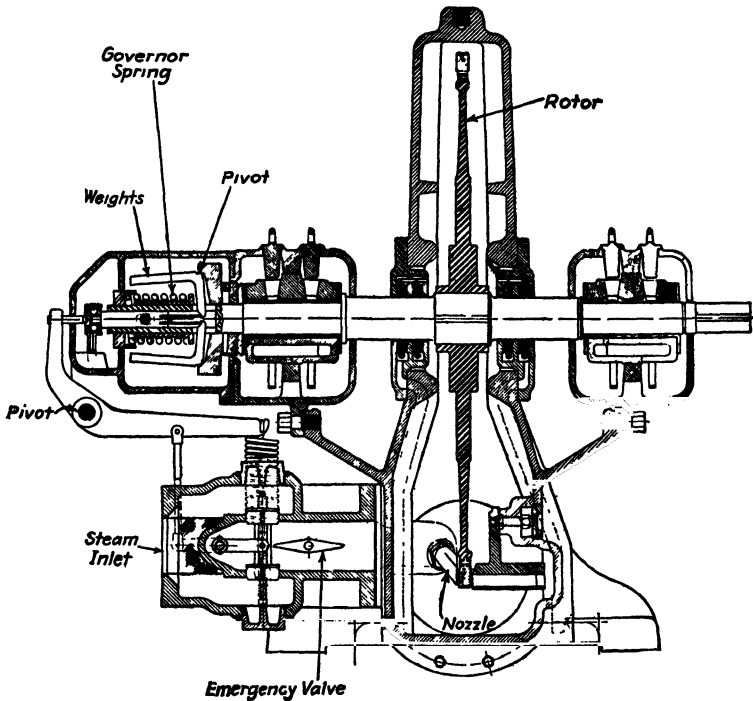



FIG. 242.—Dean-Hill single-stage turbine.

cause the governor to close the high-pressure valve, first; and, with an increase of load, the reverse will occur.

246. Back-pressure Turbines.—Turbines of this class are similar to the non-condensing turbines, except that they are equipped with means of regulation to provide exhaust steam at a definite and constant back pressure.

247. Turbine Governors.—As in the case of reciprocating engines, the purpose of the steam-turbine governor is to regulate the supply of steam to obtain constant speed under variable loads. Steam turbines are inherently a constant-speed type of engine. Their governors may act on the steam in the following different ways: (1) by

throttling the steam entering the steam chest, (2) opening or closing nozzles, (3) admitting steam to the steam chest in "puffs," the duration of each puff being changed by the governor. In addition a by-pass may be provided, by means of which high-pressure steam may be admitted to intermediate stages of the turbine. The by-pass valve is controlled by the governor and is opened when the load becomes higher than the average load for which the turbine is designed.

The turbine is, in most cases, a high-speed machine, in which there is the possibility of the rotor attaining dangerous speeds if the governor fails to operate. Therefore, an emergency governor is always provided. This automatically shuts off the steam when the speed increases above a certain predetermined limit.

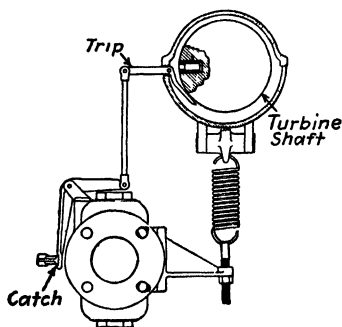



FIG. 243.—Emergency governor, Dean-Hill turbine.

A throttling type of governor which is used on practically all single-stage impulse turbines, also on many compound turbines, is shown on the turbine illustrated in Fig. 242. A centrifugal governor is enclosed in a casing and is mounted on the end of the rotor shaft. The governor weights rotate at the same speed as the shaft and are moved outward by centrifugal force. This force is opposed by an adjustable tension spring. The governor weights rest on knife-edge pivots, and their movement is transmitted to a sleeve, through a ball-bearing thrust to a series of levers which operate a double-seated, balanced, throttle valve.

Figure 243 shows the details of the emergency governor used on this turbine. A weighted plunger is held in a recess in the turbine shaft by a spring, as shown. If the normal speed is exceeded by 14 to 20 per cent, the centrifugal force throws the plunger out to where it strikes a trip lever which releases a latch on the butterfly valve, thus closing the butterfly valve and stopping the steam flow.

This Dean-Hill turbine governor is similar in design to the governor used by the De Laval, Terry and Kerr turbines (see previous figures).

A throttling type of governor used on Allis-Chalmers turbines is illustrated in Fig. 244. This governor is provided with a by-pass valve for overloads. The main and by-pass valves are operated by oil under pressure. The governor weights rotate around a vertical auxiliary shaft driven from the main shaft by gearing. Centrifugal force moves the weights outward against the spring force, and this motion is transmitted to the governor lever. This changes the posi-

tion of the oil relay valve, releasing the pressure in one end of the operating cylinder and admitting oil under pressure to the other end. This moves the operating piston and the main steam valve and, at the same time by means of the compensating levers, moves the oil relay valve back to the neutral position, closing both ports to the operating cylinder. The relay piston does not remain open but closes after the steam valve has assumed a new position.

Synchronizing of two turbines is achieved by means of a handwheel which changes the position of the relay valve with reference to

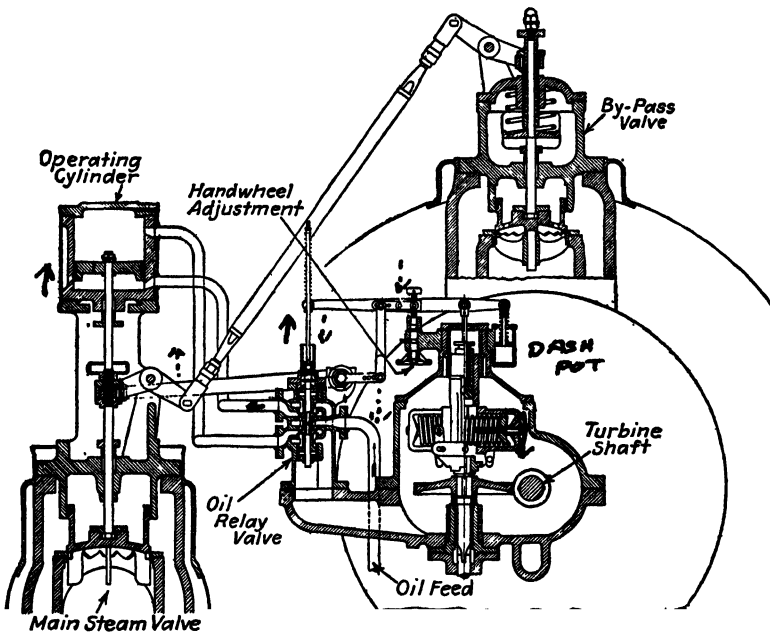



FIG. 244.—Allis-Chalmers turbine governor.

the governor. The speed can be changed about 5 per cent by this device.

As the main steam valve opens, the linkage connecting with the by-pass valve moves slightly. When the main valve is opened completely, this linkage engages the by-pass valve, and with increased load, the by-pass valve is opened. High-pressure steam is thus admitted to the intermediate stages of the turbine. A spring in the by-pass valve bonnet keeps the valve closed when not in operation. The by-pass valve permits the turbine to develop greater power. This type of governor is used on many makes of turbines, both of the impulse and reaction types.

The "Schenectady" type (Fig. 245) or hydraulic relay governor is a type of governor used on many turbines built by the General Electric Company. Its chief features are the successive opening or closing of valves to steam nozzles.

In this governor the centrifugal force of the rotating weights is opposed by the spring force as in other turbine governors. The

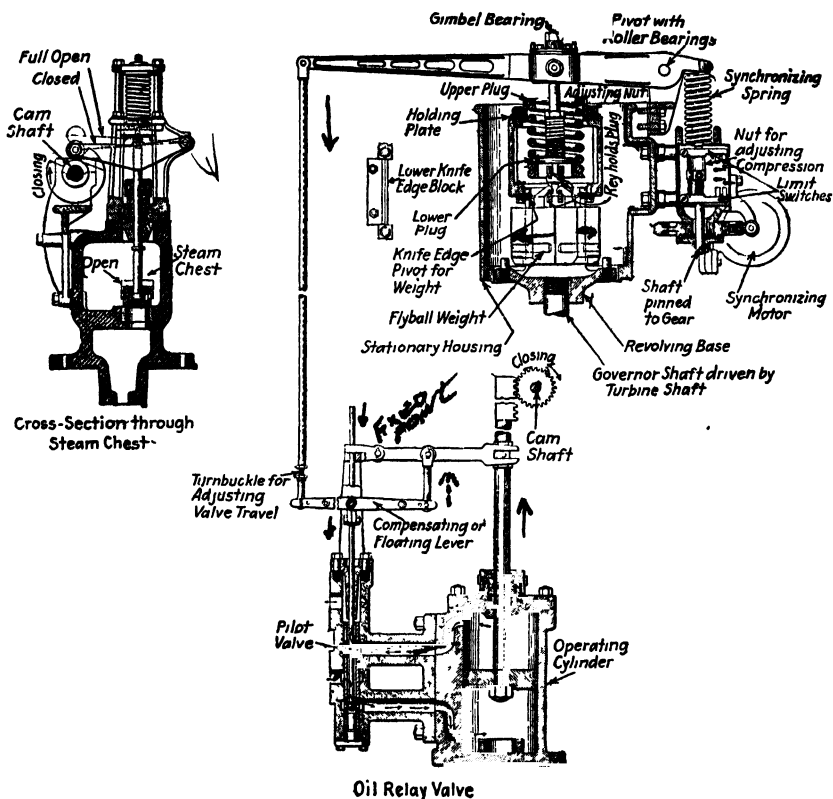



FIG. 245.—Details of "Schenectady" turbine governor.

governor lever is pivoted on roller bearings near the right end, as shown in the figure, so that its weight, acting downward, is supported by the governor spring. The synchronizing spring is under compression and this adds to the downward force on the governor spring. The synchronizing spring is used in making small speed adjustments, usually 4 per cent above or below normal, in order to properly handle the load of alternating-current generators which are to operate in parallel. It is operated by a small reversible electric motor, as shown. With an increase of speed the governor weights move out, pulling

the governor stem and lever downward, which, in turn, acts on the oil relay valve which operates the main steam valves on top of the turbine.

As seen in Fig. 245, the governor lever connects with the floating lever on the oil relay valve. As the governor lever moves down the movement is transmitted to the relay valve, admitting oil at 100 lb. per square inch pressure to the bottom of the operating cylinder and opening the top of the cylinder to the drain of the oil reservoir. Owing to the difference in pressure, the piston moves upward. By means of a pinion and rack the cam shaft is turned. The upward motion of the piston rod of the operating piston operates the floating lever to

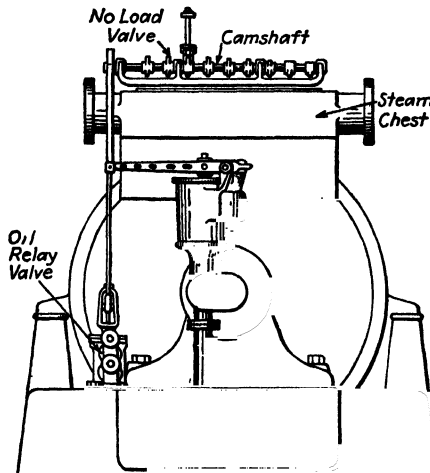



Fig. 246.—Showing assembly of Schenectady governor.

return the pilot valve to a stable position, thus stopping the upward travel.

The main-valve cam shaft is located above and carries a series of cams which operate the steam valves in succession. In operation, the events discussed take place more or less simultaneously. Figure 246 shows, in general, the assembly of the Schenectady governor.

Another type of governor used on General Electric turbines is shown in Fig. 247. Like most governors, this one is actuated by revolving weights, rotated by the main turbine shaft. In operation, the weights operate the primary pilot piston valve, which is attached to the speed governor. As the piston valve moves downward, the oil pressure under the operating piston of the primary relay is released. This action causes a movement of the system of levers which are pivoted at the points *A* and *B*, and this, in turn, causes a downward movement of the pilot valve *C*, of the secondary relay.

The downward movement of this pilot piston valve releases the oil pressure under the secondary operating piston, which, in turn, acting through a system of racks and pinions, causes a downward movement of the operating valves of the turbine. A decrease in speed causes a reverse of this action. It should be noted that a movement of the operating piston, in both the primary and secondary relays, reestablishes the pilot valve in the original position, until another impulse is received from the governor weights. A handwheel shown at the top of the primary relay valve may be used, while the turbine is in operation, to vary the speed. A movement of the handwheel changes the relative position between the primary pilot valve and bushing.

The six main steam valves are of the poppet type, and all lead to the nozzle steam chest of the first stage. The valve stems are of different lengths, so that the valves open and close consecutively.

248. Cycles of Operation.—The effect of modern turbine design toward improving the plant economy may be shown by the use of the reheat cycle which is illustrated in Fig. 248. Steam, at high pressure and temperature, is expanded, as shown, to a pressure at which it becomes saturated. It is then taken from the turbine and resuperheated, in either a fired- or steam-type reheater, after which it is returned to the low-pressure cylinder of the turbine where it is expanded to the final exhaust pressure.

In the regenerative cycle, the extraction type of turbine is used. Steam is bled from various stages of the turbine and used for heating feedwater for the boiler. At the Philo station, 28.8 per cent of all the steam entering the turbine is bled and used in evaporators and heaters.

249. Cycle Efficiency.—The actual thermal efficiency of the steam turbine is the ratio of the output to the input, both expressed, usually, in B.t.u. per hour. The method of expression is the same as in the case of the actual thermal efficiency of the reciprocating steam engine. The turbine may be considered alone, but it is more common to consider the turbine and the driven equipment together, as a unit, especially when a generator is attached. In this case the unit is called a turbogenerator, the output is in electrical units of power, and the input is the heat energy of the steam. The actual thermal efficiency may be expressed as follows (output in kilowatts):

$$e_t = \frac{3{,}414}{w(h_1 - h_{f2})} \tag{148}$$

in which

w = steam rate, lb. per kilowatt-hour.

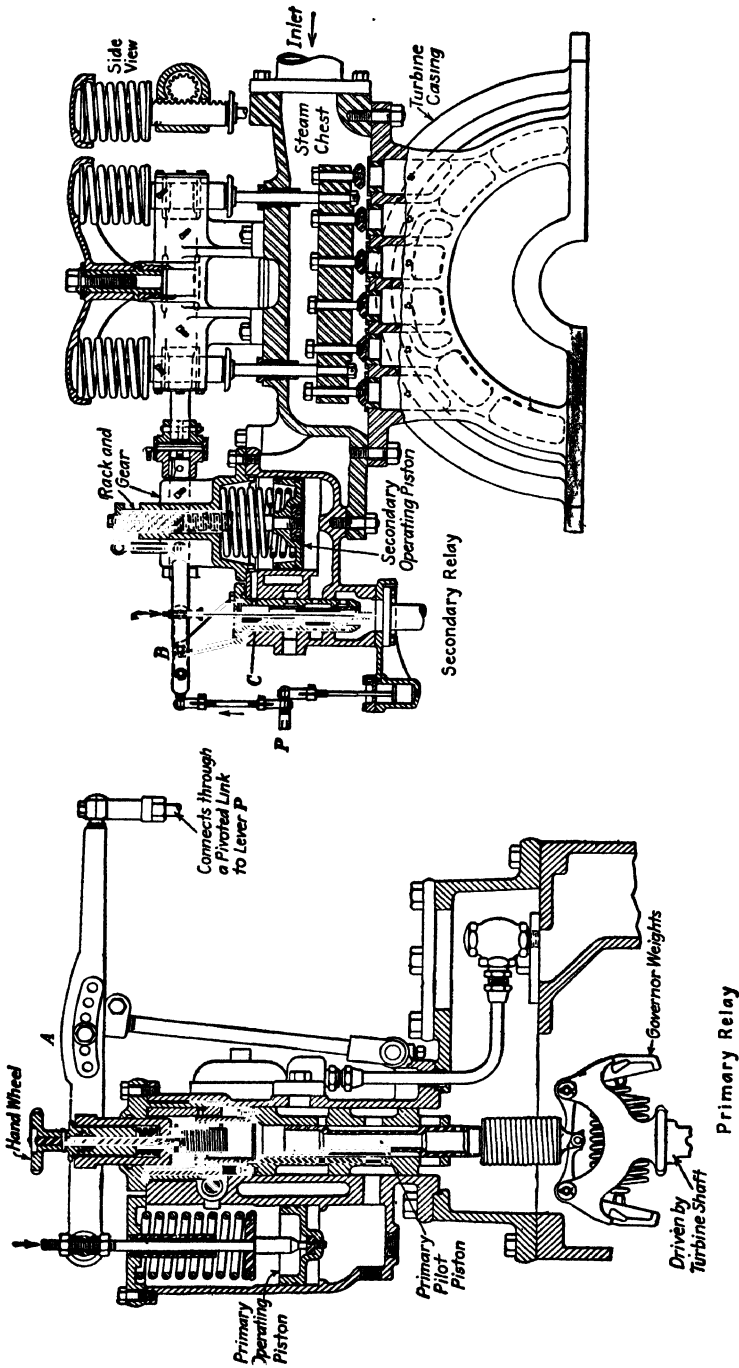



FIG. 247.—General Electric lift-bar governor.

h_1 = initial enthalpy per pound of steam, B.t.u.
h_{f2} = enthalpy of liquid, per pound at exhaust pressure, B.t.u.
3,414 = heat equivalent of 1 kw.-hr., B.t.u.

The cycle efficiency depends upon the theoretical cycle followed by the turbine. The simplest is the Rankine cycle, which was explained under steam engines (Fig. 183). The expression for the Rankine cycle efficiency is as follows:

$$e_R = \frac{(h_1 - h_2)}{(h_1 - h_{f2})} \tag{149}$$

in which

h_2 = enthalpy of 1 lb. of steam after constant-entropy expansion from p_1 to p_2.

Other symbols are as given above.

250. The Reheat Cycle.—The reheat cycle (Fig. 248) differs from the Rankine cycle in that there are at least two constant-entropy heat

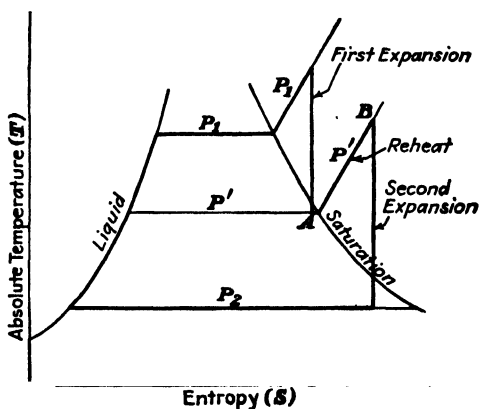



FIG. 248.—Showing reheat cycle on the T-S plane.

drops, and heat is supplied at various points in the cycle: (1) in the boiler and (2) in the reheater. The reheat cycle provides superheated steam through practically all of the turbine blading, and this greatly reduces friction and blade erosion.

The cycle efficiency for the reheat cycle is expressed in the following:

$$e_t = \frac{(h_1 - h_a') + (h_b' - h_2)}{(h_1 - h_{f2}) + (h_b' - h_a')} \tag{150}$$

in which

h_a' = enthalpy, B.t.u. per pound, after constant-entropy expansion from initial conditions to p', the reheat pressure.

h_b' = enthalpy, B.t.u. per pound, after reheating at constant pressure, p'.

h_2 = enthalpy, B.t.u. per pound, after constant-entropy expansion from reheat conditions to exhaust pressure.

h_1, h_{f2} are as before.

251. The Regenerative Cycle.—D. H. Shenk's analysis of the regenerative cycle is based on the impracticability of depicting any steam cycle by a two-dimension drawing such as have been used in the

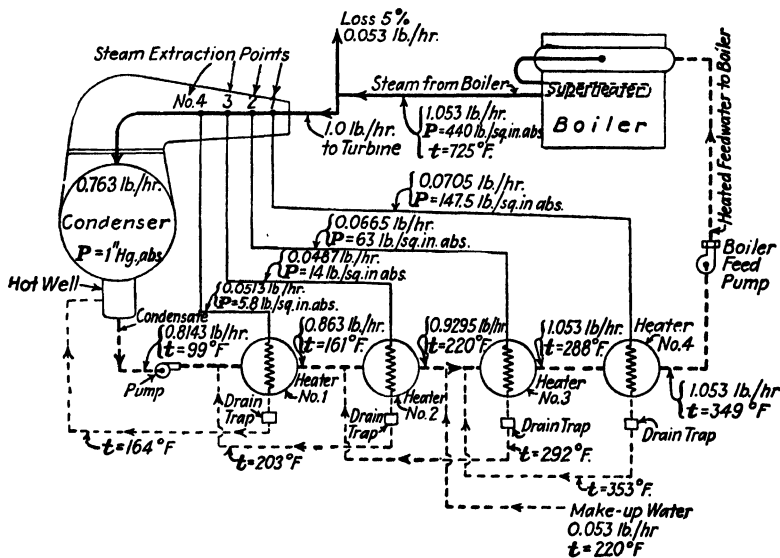



FIG. 249.—Typical regenerative-cycle diagram.

preceding discussions. The important dimensions are temperature, entropy and weight. Since the weight is a constant in most cycles it is not generally emphasized. In the case of the regenerative cycle the weight is a variable and should be considered as in the following example (Ex. 12-6).

The temperature-entropy diagram appears the same as for the Rankine cycle (Fig. 183). In the Rankine cycle all of the steam goes through all of the turbine stages. In the regenerative cycle all of the steam goes through the first few stages, but, at the first bleeder point, sufficient weight of steam at proper temperature is taken from the turbine to a feedwater heater to raise the feedwater temperature. The remainder continues in the turbine to the next bleeder point, where more steam is extracted to serve another lower temperature feedwater heater. The remaining weight of steam is expanded to exhaust conditions and discharged to the condenser.

A typical regenerative cycle diagram is shown in Fig. 249. The turbogenerator, condenser and extraction heaters are considered as a unit. The condensate from the condenser hot well passes through the series of extraction heaters, and each heater utilizes steam from the various stages of the turbine. The condensed steam from the heating coil of each heater is returned to the main condensate line, as shown. Owing to the high temperature acquired by boiler feedwater, the cycle efficiency is increased, but the available energy of the steam passing through the turbine is thereby decreased, because of incomplete expansion. The efficiency of the regenerative cycle depends, largely, upon the plant layout, and can best be shown by the following typical example. The quantity basis for the solution of this problem is 1 lb. of steam per hour to the turbine throttle.

Example 12-6.—Required, the cycle efficiency for the regenerative cycle shown in Fig. 249, using data as given in the diagram.

Solution.—*a.* Expansion of steam in the turbine. (The path of the steam expansion is shown on the *h-S* diagram, Fig. 250.)

Steam data at	Constant-entropy heat drop	Corrected for 16.7 per cent loss
440 lb., 725°F.	$s_1 = 1.64$	$s_1 = 1.64$
	$h_1 = 1{,}373$	$h_1 = 1{,}373$
147.5 lb.	$h = 1{,}253$	$h = 1{,}273$
63 lb.	$h = 1{,}178$	$h = 1{,}210.5$
14 lb.	$h = 1{,}068$	$h = 1{,}119$
5.8 lb.	$h = 1{,}012$	$h = 1{,}072$
1 in. Hg.	$h_2 = 881$	$h_2 = 964$
	$s_2 = 1.64$	$s_2 = 1.793$

NOTE.—Based on 1 lb. of steam per hour at the turbine throttle.

b. Steam extracted at point No. 1:

Heat absorbed by the water in heater No. 4 = 1.053 (320.56 − 257.15) = 66.8 B.t.u.

Heat given up by 1 lb. steam = 1,273 − 324.74 = 948.26 B.t.u.
Steam to heat 1.053 lb. water = 66.8 ÷ 948.26 = 0.0705 lb. per hour.

c. Steam extracted at point No. 2:

Water heated = 1.053 − 0.0705 = 0.9825 lb. per hour.
Heat absorbed by water in heater No. 3 = 0.9825 (257.15 − 188.06) = 67.8 B.t.u.
Heat given up by 0.0705 lb. water from heater No. 4 = 0.0705(324.74 − 257.15) = 4.8 B.t.u.
Heat supplied by steam = 67.8 − 4.8 = 63 B.t.u.
Heat given up by 1 lb. steam = 1,210.5 − 261.26 = 949.24 B.t.u.

Steam to heat 0.9825 lb. water $\frac{63}{949.24}$ = 0.0665 lb. per hour.

d. Steam extracted at point No. 3. As above,

Steam to heat 0.863 lb. water = 0.0487 lb. per hour.

e. Steam extracted at point No. 4:

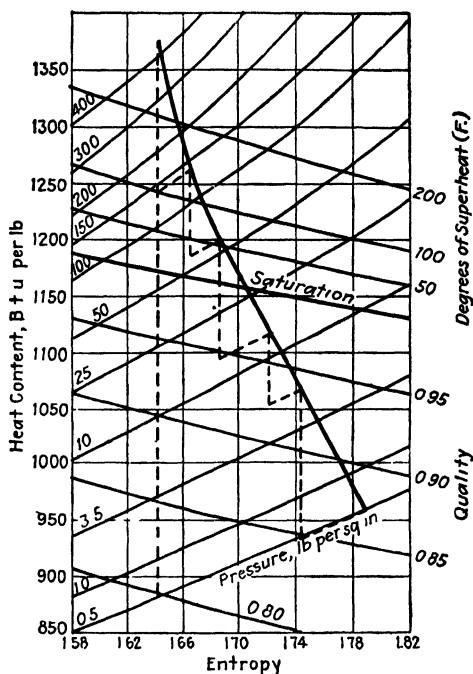



FIG. 250.—Expansion of steam in a turbine, shown on Mollier diagram.

Steam to heat 0.8143 lb water = 0 0513 lb per hour

f Steam leaving turbine at exhaust pressure of 1 in of mercury:

Steam at exhaust = 1 − (0.0705 + 0 0665 + 0 0487 + 0 0513)

= 0.763 lb. per hour.

g Available energy:

0 0705(1,373 − 1,273)	=	7 05 B.t.u.
0 0665(1,373 − 1,210 5)	=	10 80
0 0487(1,373 − 1,119)	=	12 40
0 0513(1,373 − 1,072)	=	15 40
0 763 (1,373 − 964)	=	312 00
		357 65 B.t.u.

h. Heat supplied:

1,373 − 320 56 = 1,052.44 B t u.

i. Thermal efficiency:

$$e_t = \frac{357.65}{1,052.44} = 0.34 \text{ or } 34 \text{ per cent}$$

TABLE 12-4.—WATER RATES OF TURBOGENERATORS AT VARIABLE LOADS

Rating, kw.	Initial pressure, lb. per sq. in. Abs.	Back pressure lb. per sq. in. Abs. or vacuum, in. Hg.	Superheat, °F.	Kilowatt load	Pounds steam per kw.-hr.
100	125	2 lb.	Dry saturation	25	77.8
				50	60.2
				75	50.0
				100	51.6
				125	47.2
300	175	2 lb.	100	75	56.0
				150	55.3
				225	44.3
				300	38.9
				375	35.7
5,000 cont. and max	150	28. in.	100	2,500	15.93
				3,000	15.25
				4,000	14.38
				5,000	14.41
5,000 cont.[1] 11,000 max.	185	28.5 in.	100	2,000	22.7
				3,000	20.5
				4,000	19.5
				5,000	18.9
				6,000	18.5
				7,000	18.39
				8,000	18.40
8,000 cont.[1] 16,000 max.	185	28.5 in.	100	3,000	17.3
				4,000	16.8
				5,000	16.59
				6,000	16.40
				7,000	16.28
				8,000	16.27
				9,000	16.35
20,000 cont. and max	185	28.5 in.	100	5,000	16.1
				10,000	13.4
				15,000	12.43
				20,000	12.13
				25,000	12.20
				30,000	12.50
20,000 cont. and max	185	28.5 in.	100	5,000	16.9
				10,000	14.2
				15,000	13.25
				20,000	13.70
20,000 cont. 25,000 max.	185	28.5 in.	100	6,000	15.98
				10,000	13.50
				15,000	12.75
				20,000	13.28
				25,000	13.90
27,400 cont. 30,800 hour 40,600 five min.	250	29. in.	150	27,500	11.25
				40,000	11.7
37,500 cont. 45,000 ten min.	200	28.5 in.	150	15,000	12.85
				25,000	11 43
				35,000	12.02
				37,500	12.20
				45,000	12.80
41,100 cont. 46,200 hour 60,900 five min.	250	29. in.	150	41,250	10.9
				60,000	11.5
45,000 cont. and max	185	28.5 in.	100	15,000	12.85
				25,000	11.40
				37,500	12.20
				45,000	12.80

[1] Turbines of an old type.

j. Thermal efficiency, Rankine cycle (with friction):

$$e_R = \left(\frac{1{,}373 - 964}{1{,}373 - 47.06}\right) = 0.308 = 30.8 \text{ per cent}$$

252. Steam-turbine Economy.—In discussing the economy of steam turbines, it is instructive to make a comparison of the characteristics of the turbine with those of the reciprocating steam engine. These two types of prime movers have essentially the same purpose; namely, to transform heat energy of steam into mechanical energy, which may be further transformed into electrical energy.

TABLE 12-5.—WATER RATES OF TURBOGENERATORS

Condensing				Non-condensing	
Steam at 150 lb. per sq. in. gage, exhaust, 28 in. Hg. vacuum		Steam at 200 lb. per sq. in. gage, 150° superheat, 28 in. Hg. vacuum		Steam at 150 lb. per sq. in. gage, exhaust to atmosphere	
Rating, kw.	Water rates, lb. per hr. per kw., including excitation	Rating, kw.	Water rates, lb. per hr. per kw., including excitation	Rating, kw.	Water rates, lb. per hr. per kw.
50	32.0 to 42.0	4,000	12.0 to 13.3	50	52.0 to 60
100	21.2 to 27.5	5,000	11.9 to 13.0	100	43.5 to 50
200	19.5 to 25.0	7,500	11.6 to 12.6	200	38.4 to 44
300	18.7 to 22.3	10,000	11.2 to 12.3	300	36.5 to 42
400	18.2 to 21.0	12,500	11.1 to 12.0	400	35.2 to 40
500	17.7 to 19.3	15,000	11.1 to 11.8	500	34.1 to 38
600	17.4 to 19.0	17,500	11.0 to 11.6	600	33.4 to 37
750	17.0 to 18.8	20,000	11.0 to 11.4	750	32.7 to 36
1,000	16.5 to 18.6	25,000	10.9 to 11.3	1,000	31.7 to 35
1,250	16.2 to 18.3	30,000	10.8 to 11.2	1,250	31.0 to 34
1,500	16.0 to 18.0	35,000	10.7 to 11.1	1,500	30.7 to 33
2,000	15.7 to 17.7	40,000	10.5 to 11.1	2,000	30.1 to 32
2,500	15.5 to 17.4	50,000	10.5 to 11.0	2,500	29.4 to 31
3,000	15.4 to 17.1	60,000	10.5 to 10.9	3,000	28.9 to 30
3,500	15.3 to 16.9	75,000	10.4 to 10.8	3,500	28.6 to 29

The reciprocating steam engine is superior to the turbine for cases where variable speeds are required, and also for slow speeds combined with a heavy starting torque. The unaflow engine gives a remarkably low water rate and, because of its simple and almost frictionless valve gear, has a very high mechanical efficiency. As a result, the unaflow engine shows superior economy in sizes of less than, 1,000 hp. The steam turbine, however, has certain fundamental advantages over the reciprocating engine, the principal ones of which are:

1. Low first cost.

3. Lighter foundations.
4. Less operating attendance.
5. Better utilization of high vacuum.
6. Lower oil consumption.
7. Less vibration.
8. Capable of handling high overloads.
9. Elimination of oil from exhaust steam.
10. Close speed regulation.
11. Capable of using steam at high temperatures.
12. High speed desirable for driving electrical equipment.

The results of steam-turbine tests as given by the American Railway Association are shown in Tables 12-4 and 12-5.

For the best efficiency, high steam pressures and temperatures and high vacuums should be used. The large turbines show excellent economy because their design includes refinements and effective auxiliary equipment.

Problems

1. Determine the dimensions of a nozzle for (*a*) theoretical conditions, (*b*) friction loss calculated by Eq. (138), using the following data: $p_1 = 220$ lb. per square inch absolute; $t_1 = 590°F.$; $p_2 = 20$ lb. per square inch absolute; steam flow is 7,500 lb. per hour; $p_0/p_1 = 0.55$. (Note.—In part *b* the friction loss affects the diameter at the mouth only.)

2. A turbine with four pressure stages is supplied with steam at a pressure of 185 lb. per square inch absolute; 90° of superheat; exhausting at 5.5 lb. per square inch absolute; speed, 1,200 r.p.m. The axis of the nozzle is set at 20 deg. with the direction of the blade travel.

For theoretical conditions, determine: (*a*) steam-jet velocity leaving the nozzle, (*b*) blade velocity, (*c*) velocity of steam leaving the blades, and blade efficiency, (*d*) diameter of the rotor, or mean blade diameter, (*e*) Rankine cycle efficiency, (*f*) pressure, temperature and quality of steam leaving each stage.

3. Find the results of Problem 2, friction loss 20 per cent.

4. A turbine uses 12.4 lb. of steam per kilowatt-hour; steam pressure 290 lb. per square inch absolute; temperature 550°F.; and exhausting at 6 in. of mercury absolute. Determine the thermal efficiency, Rankine cycle efficiency, and the Rankine cycle ratio.

5. A four-pressure-stage turbine operates under the following conditions; steam pressure 265 lb. per square inch absolute; quality 0.985; exhausting at 10 in. of mercury absolute; mean blade diameter 48 in. Assuming theoretically perfect conditions, determine (*a*) the steam-jet velocity, (*b*) blade velocity, (*c*) speed in r.p.m. and (*d*) the pressure and quality of the steam leaving each stage.

6. Solve Problem 5, friction loss 16.7 per cent.

7. A 750-kw., impulse turbine with four pressure stages is designed for equal heat drop in each stage. Data as follows: p_1, 325 lb. per square inch absolute; t_1, 625°F.; p_2, 1.4 lb. per square inch absolute. (*a*) Calculate the steam velocity and theoretical mean blade diameter for a speed of 3,600 r.p.m. (*b*) Determine the pressure and quality or temperature in each stage. Assume theoretical expansion.

8. A nozzle is to deliver 1 lb. of steam per second from a pressure of 200 lb. per square inch absolute, and quality, 0.985 to an exhaust pressure of 14.7 lb. per square inch absolute. The angle between the steam jet and the direction of motion of the blades is to be 20 deg. (*a*) Determine the diameters, in inches, of the throat and mouth of the nozzle; also the length, in inches. (*b*) Determine the horsepower imparted to the rotor.

9. A nozzle is to expand 2,500 lb. of dry steam per hour from a pressure of 210 lb. per square inch absolute to 5 lb. per square inch absolute. Find the diameter of the throat and mouth, and the length based on a divergence angle of 6 deg. between the sides and axis of the nozzle. Dimensions to be in inches.

10. A nozzle is to deliver 30 lb. of steam per minute from an initial pressure of 200 lb. per square inch absolute, and 50° of superheat, to a vacuum of 12.6 in. of mercury. Barometer is 29.4 in. of mercury. Friction loss calculated by Eq. (138). Determine the diameter at the throat and end of the nozzle. Dimensions to be in inches.

11. Determine the theoretical thermal efficiency of a steam plant operating on the reheat cycle, when steam at 500 lb. per square inch absolute and 700°F. expands to 90 lb. per square inch absolute, and then is reheated at constant pressure to a temperature of 650°F. From this point it expands to the exhaust pressure of 2 in. of mercury absolute. Expansion takes place at constant entropy.

12. Design a nozzle to expand 2,000 lb. of steam per hour from an initial pressure of 250 lb. per square inch absolute to a back pressure of 20 lb. per square inch absolute for steam that is (1) dry and saturated, (2) superheated 50 degrees, (3) wet with a quality of 0.978. The expansion is at constant entropy in each case. Dimensions to be in inches.

13. Solve Problem 12, friction loss calculated by Eq. (138).

14. A plant operating on the reheat cycle uses steam at 400 lb. per square inch absolute, and 200° of superheat. The steam is expanded until the quality becomes 98 per cent, and then a constant pressure reheating brings the temperature to 500°F. Exhaust pressure is 5 lb. per square inch absolute. Assuming that the expansions are at constant entropy, determine the cycle efficiency.

15. In a plant operating on the regenerative cycle, steam goes to the turbine at 350 lb. per square inch absolute and with 200° of superheat. Steam is bled from the turbine at 60 and 13 lb. per square inch absolute. The exhaust pressure is 2 lb. per square inch absolute. The water temperatures are as follows: leaving the condenser, 120°F.; leaving the low-pressure heater, 195°F.; leaving the high-pressure heater, 285°F. Steam condensed in the high-pressure heater enters the low-pressure heater, and the condensate from the low-pressure heater enters the condenser. The temperature of the condensate leaving each heater is 5° less than the saturation temperature of the steam entering the heater. Draw a diagram and calculate the thermal efficiency, assuming adiabatic expansion.

16. Solve Problem 15, assuming that there is a 17.5 per cent friction loss in the turbine.

17. A plant operating on the regenerative cycle supplies steam to a turbine at 400 lb. per square inch absolute, and 700°F. Exhaust pressure is 1 in. of mercury. Steam is bled at two points: at 49.7 and 14.4 lb. per square inch absolute pressure. The condensate from the high-pressure heater is at 270°F., and enters the low-pressure heater. The condensate from the low-pressure heater is at 202.5°F. and enters the condenser. The water temperatures are as follows: leaving the condenser, 77°F.; leaving the low-pressure heater, 202.5°F.; and leaving the high-pressure heater, 267°F. Determine the thermal efficiency, assuming a 23 per cent friction loss in the turbine.

CHAPTER XIII

CONDENSING EQUIPMENT

253. Introductory.—The method of producing a vacuum by condensing steam in a confined chamber has been understood by engineers since the first occurrence of reciprocating steam engines. Newcomen's engine was essentially a "vacuum" engine; one in which the steam was admitted into the vertical cylinder, beneath the piston, at atmospheric pressure. By spraying water into the cylinder, the steam therein was condensed. A vacuum resulted, and with the atmospheric pressure acting on the upper side of the piston, it was forced downward, giving a complete power stroke. James Watt improved the steam economy of Newcomen's engine by using a condenser separate from the cylinder.

254. Principle of Condenser Operation.—The principle involved in the operation of a condenser can be explained by means of the following simple example:

If 100 lb. of dry, saturated steam at an absolute pressure of 14.7 lb. per square inch were contained in a closed tank, the volume required would be $100 \times 26.82 = 2{,}682$ cu. ft. Now, if the steam is condensed to water, the resulting volume of the water would be $100 \times 0.016 = 1.6$ cu. ft. After the cooling, the tank with a volume of 2,682 cu. ft. contains only 1.6 cu. ft. of water. This would indicate that a high vacuum exists in the tank.

The practical problem is, however, not so simple as in the example given, the chief reason being that the material handled by the condenser is not water or steam, alone, but a mixture of air and water vapor. The air reaches the steam through leaks in the vacuum system, and, also, by way of the boiler feedwater.

According to the law of partial pressures, the total pressure within a condenser is the sum of the partial pressure of the air and water vapor. If the condenser contains only water and its vapor, and the temperature is assumed to be 106°F., the absolute pressure would be 2.3 in. of mercury, or a vacuum of 27.7 in. of mercury, based on a 30-in. barometer. For example: it will be assumed that 0.2 lb. of air is mixed with every pound of steam in the condenser. The volume occupied by the 0.2 lb. of air will be the same as that occupied by the saturated steam with which it is mixed. or 296.6 cu. ft. The tempera-

ture of the air is the same as that of the steam. Then, the pressure of the air is:

$$P = \frac{WRT}{V} = \frac{0.2 \times 53.35 \times 566}{296.6} = 20.35 \text{ lb. per square foot}$$

$$p = \frac{20.35}{144 \times 0.491} = 0.29 \text{ in. of mercury}$$

Then, the total pressure in the condenser will be 2.3 + 0.29 = 2.59 in. of mercury or a vacuum of 27.41 in. of mercury.

With 1 lb. of air per pound of steam (a possible condition), the vacuum would be 26.26 in. of mercury.

Thus, air in moderate amounts reduces the vacuum in condensers, and if not continuously removed would gradually increase the pressure and impair the efficiency of the prime mover exhausting into it. If a high vacuum is desired, the condenser should be provided with equipment for cooling the air, compressing and discharging it to the atmosphere. In some cases, a condenser is used primarily for retaining the condensate that it may be returned to the boiler as feedwater. In such cases there is no means for maintaining a high vacuum.

255. Classification.—Condensing equipment includes the condenser proper, and all auxiliary equipment necessary for the circulation of the cooling water and removal of condensate and non-condensable gases.

Condensers are generally classified in the following manner:

1. Jet condensers.
   a. Low-level type.
   b. Barometric type.
   c. Ejector type.
2. Surface condenser.

Jet condensers attempt to bring the cooling water into intimate contact with the steam to be condensed. The resulting mixture of condensate and cooling water is usually discharged from the condenser at atmospheric pressure. The condensate from this type of condenser is generally not suitable for boiler feed, unless the cooling water is of high quality.

Jet condensers are built to operate with (1) parallel flow, in which the steam, entrained gases and cooling water enter at the top of the condenser and flow downward together, or (2) counterflow, in which the cooling water enters at the top and flows downward, while the steam enters the lower part of the condenser and flows upward, the entrained gases being removed at the top.

A *surface condenser* consists of a large metal chamber with suitable steam and water connections. The exhaust steam from an engine or turbine enters and passes among tubes through which cooling water flows. A single-pass condenser is one in which the cooling water flows from one end to the other, and a multi-pass condenser is one in which the water flow is reversed through the steam chamber one or more times. In the usual design of surface condenser, the cold water enters the first pass at the bottom, and returns through a second pass at the top. The exhaust steam enters at the top and the condensate leaves at the bottom; hence, the flow is countercurrent.

The auxiliary equipment with a surface condenser includes a circulating pump for cooling water, a pump for removing air and non-condensable gases, and a pump for discharging the condensate.

Large-size surface condensers are supported on four brackets at the base, which rest on coil springs. This arrangement relieves the turbine frame from the enormous weight of the condenser and the contained water. In some cases, jacks are provided to change the force of the springs in keeping the weight on the turbine flange constant.

In comparing the two types of condensers, the principal advantages of the jet condenser are: (1) simplicity of design, (2) lower first cost, (3) lower upkeep and (4) small floor space required. Under bad water conditions the low cost of upkeep is an important item. Acid water such as is found in some coal-mining districts will corrode the tubes of a surface condenser so fast that the maintenance expense is prohibitive. Jet condensers are often used in stand-by stations because the cost of cleaning the boilers is more than balanced by the increased fixed charges that would occur on a surface condenser which is idle part of the time.

The principal advantages of the surface condenser are: (1) condensate may be utilized by returning it to the boiler, eliminating expense of treating all of boiler feedwater, (2) less air is carried to boiler, and (3) auxiliary power required is less than with jet condenser. There are certain localities where pure water is abundant and can be fed into the boiler without expensive chemical treatment. In the usual plant, however, the raw water available contains scale-forming impurities and other foreign matter. All such water must be purified before entering the boiler. This factor, alone, is often the deciding element in favor of the surface condenser.

256. Low-level Jet Condensers.—Of the three types of jet condensers the low-level type is probably the most commonly used.

Figure 251 shows a sectional view of a low-level jet condenser with two condensate pumps and a hydraulic air pump. The cooling

water enters the condenser proper through the annular chamber surrounding the top of the condenser head. Equally spaced around the periphery of the steam chamber are nozzles containing spiral vanes. Through these nozzles the water is discharged and directed toward the center of the chamber, in a whirling spray. The exhaust steam enters at the top of the condenser and passes downward, through this water spray, effecting intimate contact, and resulting in condensation of the steam.

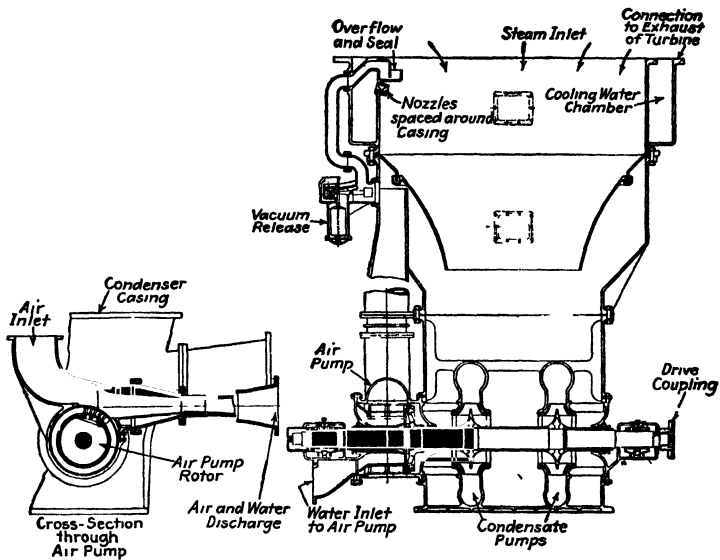



FIG. 251.—Westinghouse jet condenser.

At the left of the condenser head, as shown, and connected to the upper part of the casing through a water seal, is the vacuum breaker, a protective device which prevents water from flooding the main turbine in case the condensate pumps fail. If water rises in the head it overflows into the vacuum breaker line and float chamber. This causes the float to rise, opening a valve to the atmosphere, and thus breaking the vacuum. In this event the atmosphere relief valve in the exhaust steam line would open, and the turbine or engine would continue to operate, non-condensing.

Air and non-condensable gases are removed by a Leblanc hydraulic air pump. The air pump is connected to the condenser chamber, under the conical hood, where these gases have been thoroughly cooled.

Figure 252 shows a low-level jet condenser with duplicate condensate pumps and using air ejectors in place of a hydraulic air pump for removing non-condensable gases.

If the suction head for the injection water is not too high, the water may be siphoned into the condenser by the vacuum. About 18 ft. is considered as the maximum lift allowable. With lifts of 20 ft. or more, there may be the danger that an increase in the load will decrease the vacuum so that insufficient injection water would be drawn in to continue operation. A pump or some other priming device must be used in starting for this sort of operation.

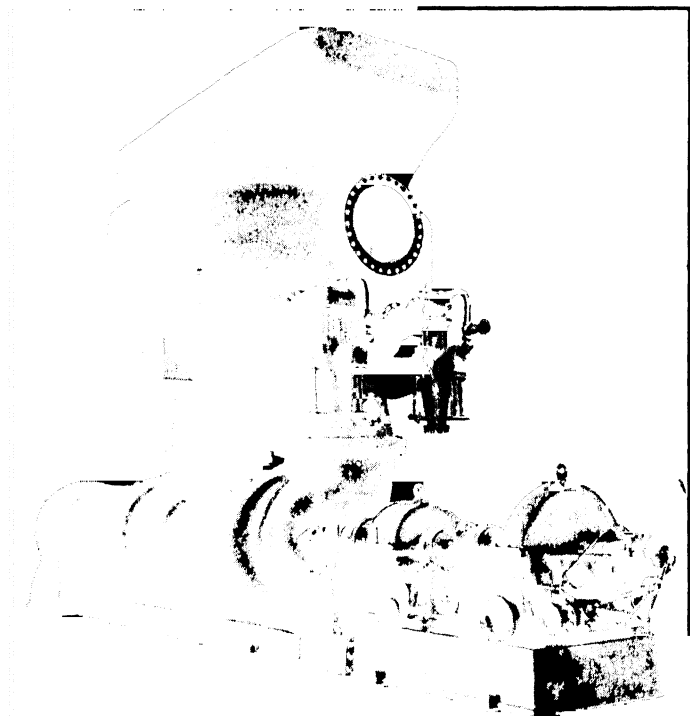



Fig. 252.—Wheeler jet condenser with steam-jet air pumps Capacity, 15,000 kw.

257. Barometric Jet Condenser.—It may be seen in Figs. 253 and 254 that the barometric jet condenser is similar to the low-level jet condenser except that it uses a barometric tube or tail pipe for creating the vacuum and removing the condensate, and, in some cases, the non-condensable gases. The low-level condenser is small in overall height, but the barometric condenser requires about 34 ft. above its base. The reason for this is that the tail pipe must be long enough to provide a barometric column of water and also some space above for a condensing chamber.

The barometric condenser is most suitable when it is to serve equipment situated at an elevation so that the condenser head may

be placed below the turbine. For high vacuum, some form of air pump is necessary. Where the condenser is above the turbine, it is necessary to use a long exhaust pipe, which increases the cost of the installation and causes increased resistance to the steam flow.

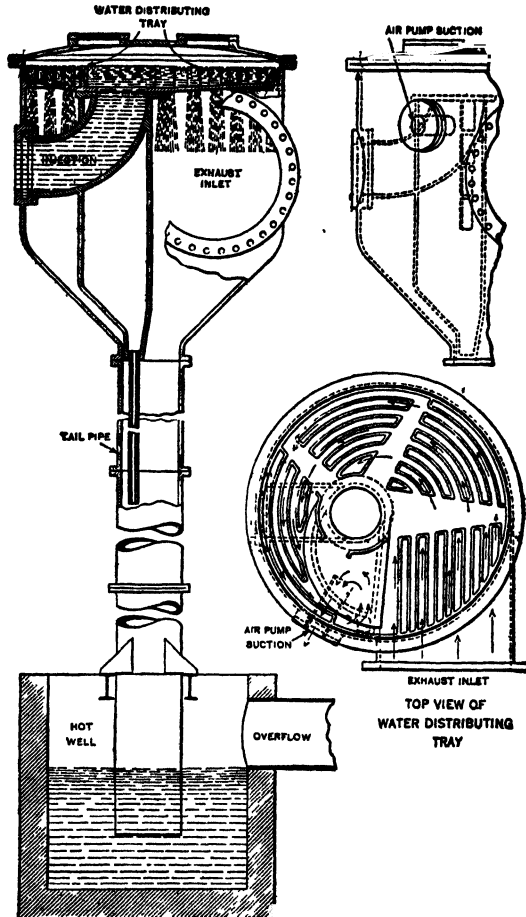



FIG. 253.—Wheeler barometric condenser.

Figure 253 shows a typical barometric condenser in which the injection water is introduced into distributing trays at the top. From here it spills, in thin sheets, through the condensing chamber, where it comes in contact with the exhaust steam. The water and condensate drop, by gravity, into the tail pipe, in which the water level is at a definite height, depending on the vacuum in the condenser. The entrained air is drawn upward into an air chamber and discharged to the atmosphere by an air pump. The injection water may be drawn

in by the vacuum if the lift from the supply is not more than approximately 18 ft. Otherwise, it may be supplied by a pump. An atmospheric relief valve may be provided. The water resulting from the condensing process is removed from the condenser by gravity and this eliminates pump troubles often experienced in the handling of hot water. Any variation in the condenser vacuum is automatically cared for by a rise or fall of the water level in the tail pipe.

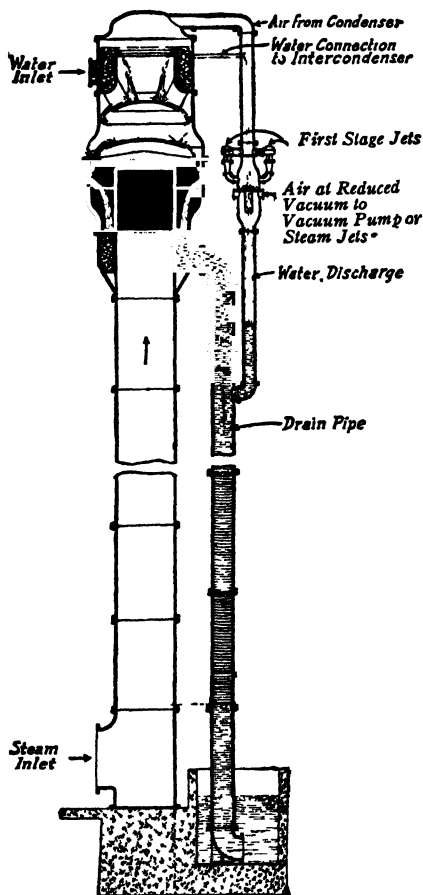



Fig. 254.—Ingersoll-Rand barometric condenser with central inlet.

In Fig. 254 is illustrated a barometric condenser of the countercurrent type. The condensing water, as indicated, enters an annular space near the top and spills onto an annular baffle above the steam pipe. Hence, the exhaust steam, entering the condenser chamber from below, is entirely surrounded by a sheet of water, producing effective condensation. The air released from the steam is relatively high in temperature and contains a large amount of water vapor. The mixture passes upward through the falling sheets of cold water, where much of the water vapor is removed The small amount of air is taken off at the extreme top of the casing by some sort of steam-jet vacuum equipment or hydraulic air pump.

258. Ejector Jet Condenser.—The ejector condenser operates on a principle similar to that of the steam injector often used to force feedwater into small boilers. The injection or cooling water is discharged under pressure, through one or more nozzles. Exhaust steam on entering the condenser, comes in contact with the jets of water and is condensed. The cooling water, condensate and non-condensing gases are all carried downward by the force of the stream and discharged through a suitable diffusing tube. The kinetic energy of the

water gives enough momentum to the mixture to maintain the vacuum. The pressure is increased to that of the atmosphere and the water and entrained gases flow to the discharge line.

An ejector jet condenser equipped with a Leblanc hydraulic pump is illustrated in Fig. 255. The pump discharges the cooling water into a series of cones, entraining the condensate and gases as it passes through them. In the diffuser cone the pressure is increased to that of the atmosphere.

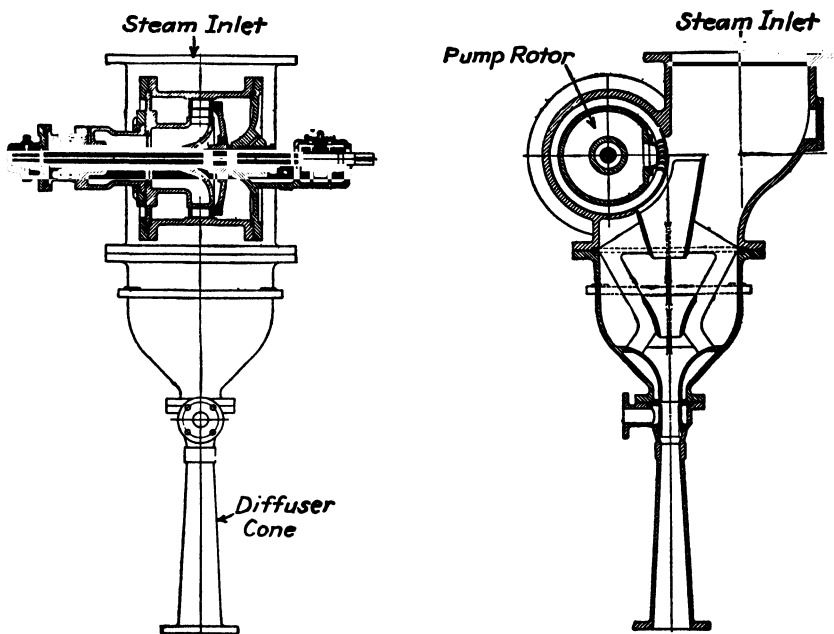



FIG. 255.—Westinghouse ejector condenser.

This type of condenser is generally used for small installations only and is not well adapted for intermittent service, nor for frequently varying loads.

Figure 256 illustrates a typical multi-jet condenser installation with auxiliary equipment. The injection water, under pressure produced by the pump, passes through a ring of nozzles, downward through the combining tube, and into the throat of the diffuser tube, where the water jets unite to form a single jet. The velocity attained by the water jets is sufficient to entrain the condensate, air and other gases and to discharge them into the hot well. The water jets create the vacuum by condensing the steam, and maintain it by removing the air and condensate. No separate vacuum pumps are required.

A float-operated vacuum breaker is provided to admit air to the vacuum chamber in case the flow of injection water fails. In event of this, the vacuum is displaced by water from the hot well, through the diffuser tube. The breaker protects the turbine or engine exhaust lines against the danger of being flooded with water.

An atmospheric exhaust valve is connected to the exhaust line from the turbine. It opens automatically when the pressure in the

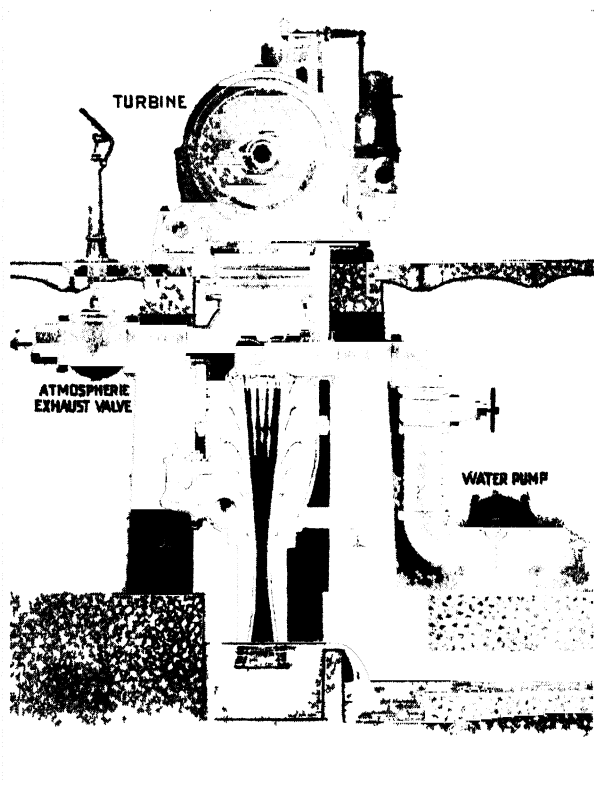



FIG 256 —Schutte-Koerting multi-jet condenser, shown under turbine.

condenser becomes greater than that of the atmosphere, and closes as soon as there is a partial vacuum in the condenser. When open, the turbine exhaust is allowed to flow to the atmosphere.

The connection shown between the turbine and condenser consists of a corrugated copper tube with cast-iron flanges. This joint permits movement due to expansion and contraction resulting from heat variation.

259. Surface Condensers.—Condensers of this type are built in various sizes, ranging, in amount of cooling surface, as high as

90,000 sq. ft. and higher. They are adaptable equally well to both reciprocating steam engines and steam turbines, but are generally given preference in cases of the latter.

Surface condensers are placed as close as possible to the equipment which they serve. The auxiliary equipment is generally installed in the immediate vicinity of the condenser, though an exception to this occurs, occasionally, in the case of the circulating-water pumps.

All surface condensers are constructed essentially as shown in Fig. 257; excepting, of course, the auxiliary equipment. The outer shell and heads are of cast iron, and the tubes and tube sheets are

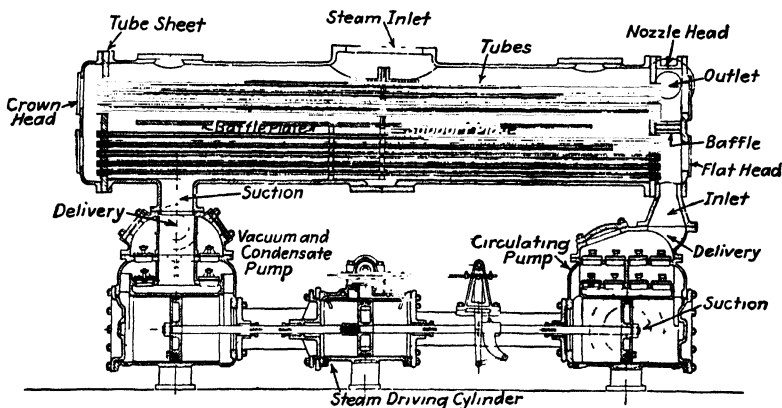



FIG. 257.—Surface condenser unit (Worthington).

generally of special copper alloy metals. In the condenser shown, the arrangement is such that the cooling water enters at the lower, right-hand end and passes through two sets of tubes, in series, before leaving at the top. The steam enters through the upper connection, as shown, and is deflected by the baffles as it flows downward, across the tubes, in the course of condensing. The condensate and non-condensable gases are removed by a single pump, from the lower, left end. Vertical and horizontal baffles are arranged in the steam chamber to assure thorough distribution of the steam.

The tubes are held in the tube sheets by suitable packing glands so as to permit their free expansion and contraction. The condenser unit illustrated in Fig. 257 is frequently used with reciprocating engines.

Large-sized, horizontal, surface condensers are built, in general, as shown in Fig. 258. This condenser, however, is distinctive in that it is designed for "radial flow" of the steam into the tube nest. This is accomplished by providing the annular space around the inner surface of the shell. The larger portion of the steam permeates this

space before entering the nest of tubes proper, and the horizontal and vertical *steam lanes* facilitate the flow to the inner portion of the nest.

The cooling water enters at the bottom, and, as in the previous example, flows through two passes of tubes before leaving the con-

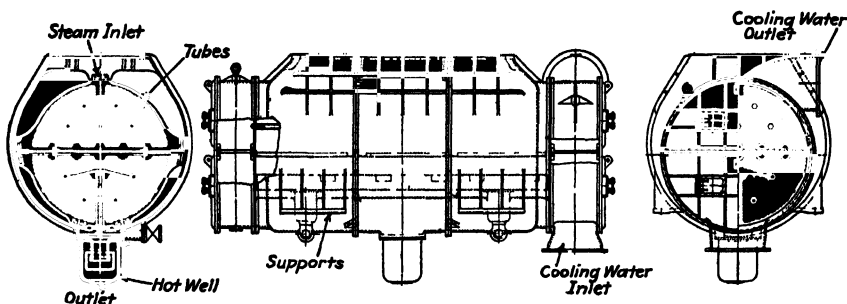



FIG. 258.—Westinghouse surface condenser.

denser. Condensate drips over the tubes, to the bottom of the shell, and is collected in the hot well, from which it is removed by a pump. The gases are removed from the lower half of the tube nest, through the connections, as shown in the section view.

Figure 259 illustrates, diagrammatically, two methods of arranging the tubes of large surface condensers to promote the flow of steam into

(*a*) Cross flow. (*b*) Radial flow.

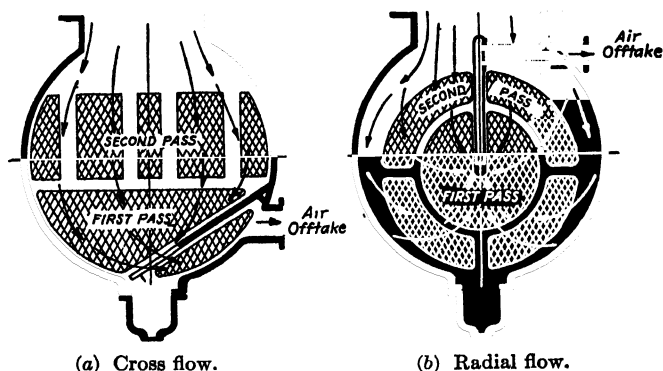



FIG. 259.—Diagrams showing methods of steam flow in surface condensers.

all parts of the tube nest. In Fig. 259 *a* the cooling water flows through the tubes, in two passes, as shown. Vertical and horizontal steam lanes, within the tube nest, are provided, and the gas outlet is at the bottom, below the coolest tubes. To a surface condenser of this type is sometimes attributed the name "downflow" or "crossflow."

In Fig. 259 *b* is illustrated a very advantageous arrangement to effect radial flow of the steam into the nest. An annular space is

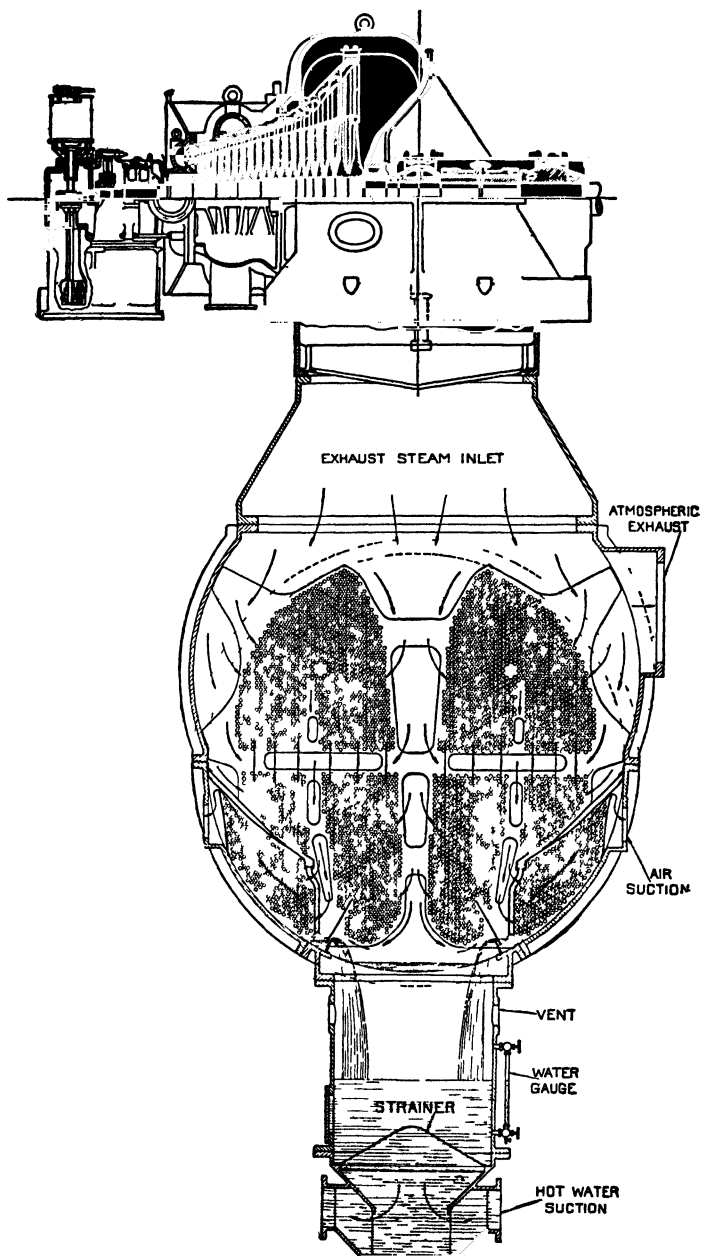



FIG 260—Worthington surface condenser, shown with an impulse turbine, to show relative sizes of turbine and condenser.

provided between the tubes and the condenser shell, similar to the arrangement shown in Fig. 258. The steam lanes and the arrangement of the cooling water passes are, however, somewhat different. The baffles in the condenser heads are arranged so as to cause the cooling water to flow through the middle of the nest on making the first pass, and around the outside on the second or return pass. This method divides the cooling surface into more or less distinctive zones, which assures effective functioning of the condenser. The gases are removed from the coolest zone, as shown. The condensate, on dripping over the hotter tubes in the lower portion of the second pass, is slightly reheated. By close regulation of the cooling water, the condensate may be removed at very near saturation temperature. Surface condensers which effect a minimum of refrigeration (cooling below saturated temperature) of the condensate have an advantage in regenerative heat-cycle operation.

Figure 260 shows the general arrangement and relative size of a large surface condenser used in connection with a steam turbine. The axis of the condenser is horizontal, and it may be arranged either at right angles or parallel with the turbine shaft.

The condenser illustrated in Fig. 260 employs a type of construction which permits cleaning the tubes in one-half of the tube nest while the other half is in operation. This is accomplished by providing a cooling-water inlet and outlet on each side, and by dividing the heads into two sections by vertical baffles. Thus, there are two passes on each side, and the head covers in one of the semicylinders may be removed and the tubes cleaned while cooling water is flowing in the other. This arrangement is often termed *double flow*.

The most common type of construction for surface condensers is that in which the incident features include a horizontal cylinder, and two, single-flow passes for the cooling water. Deviations from this design are many and varied and often include *single* or *multiple pass*, *multiple flow* and arranging the axis vertically. There are many condensers in service which have any one or all of these features incorporated in their construction.

An example of a surface condenser with a triangular-shaped shell is shown in Fig. 261. Figure 262 gives a general idea of an arrangement in an installation of vertical condensers. They are used where floor space and head room are to be kept within narrow limits. In the installation shown in Fig. 262, the exhaust from the low-pressure cylinder of the turbine divides and flows to four vertical condensers. Each condenser is constructed for single pass and upward flow of the cooling water through the tubes. For cleaning, an auxiliary pump is

used to reverse the water flow. Separate auxiliary equipment is provided for each condenser.

260. The Rand Condenser Unit.—Figure 263 illustrates, diagrammatically, the construction and operation of a single-pass surface condenser equipped with a separate air cooler. The shape of the shell and the arrangement of the tubes, within, is such as to effect flow and condensation according to the volume of steam in the various regions. Thus, the sides of the shell converge, and the steam space decreases toward the bottom.

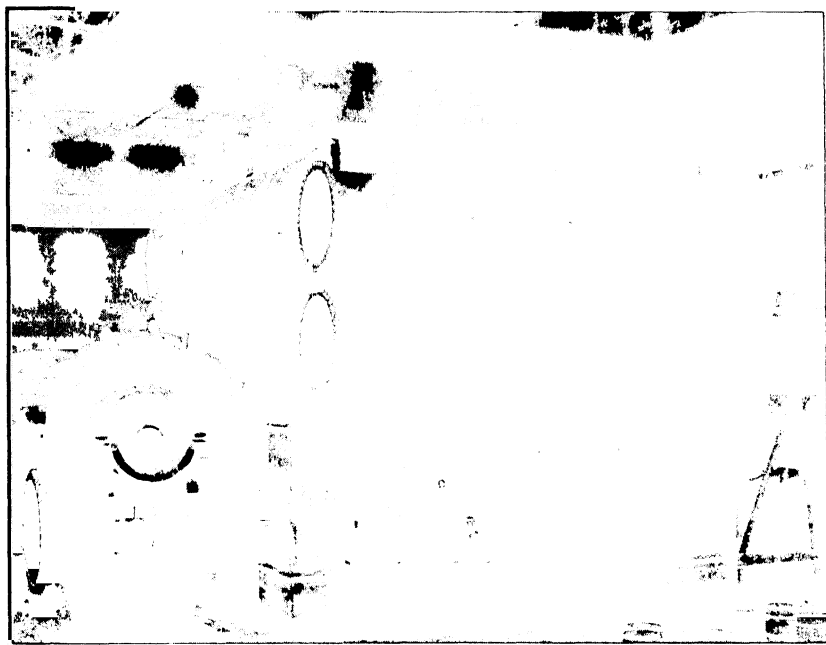



FIG 261 —Photograph of a shell for a surface condenser having 90,000 sq. ft. of cooling surface. Capacity to serve a 160,000-kw. turbogenerator (Ingersoll-Rand).

The air and water vapor are removed from the upper space of the hot well and pass into a separate air cooler and vapor condenser unit. By this means the vapor is condensed out before the air is discharged to the atmosphere. The unit is essentially a small surface condenser consisting of an external cooler, intercondenser, and an aftercondenser, as shown in the diagram of Fig. 263. Cooling water taken from the main supply line is passed through the various parts of the unit, as indicated. The air and vapor enter the cooler and are directed by the baffles through three passes along the tubes. Cooling water flows through the tubes in the same three passes, and

its direction is opposite in each pass. The water and vapor flows are *thus countercurrent*. By the time the vapor and air have traversed the entire length of the cooler passages, devaporization is completed, and the air is cooled to about the temperature of the inlet circulating water. Air leaving the cooler enters the first-stage steam-ejector jet and is discharged at a higher pressure to the intercondenser. The

FIG. 262.—Photograph of a 94,000-kw. turbogenerator unit, equipped with Ingersoll-Rand vertical surface condensers.

intercondenser, also, is constructed with three passes. In the tubes of the first two passes flows condensate from the main condenser, absorbing heat from the first stage ejector. In the third pass, circulating water cools the air before it enters the second-stage steam ejector. The second-stage steam jet discharges the devaporized and cooled air at atmospheric pressure into the aftercondenser. Condensate from the first two sections of the intercondenser passes through the tubes of the aftercondenser, where it absorbs the available heat of the steam

from the ejector. Hence, most of the available heat in the ejector steam is returned to the boiler by way of the condensate. The drains from the precooler and intercondenser are returned to the main-condenser hot well.

261. General Details of Surface-condenser Construction.—The recent trend in surface-condenser design has been toward a gradual decrease in the square feet of tube surface required per kilowatt capacity of the turbine. In a prominent central-power-plant installation this value is 0.562 sq. ft. per kilowatt. Factors which influence this

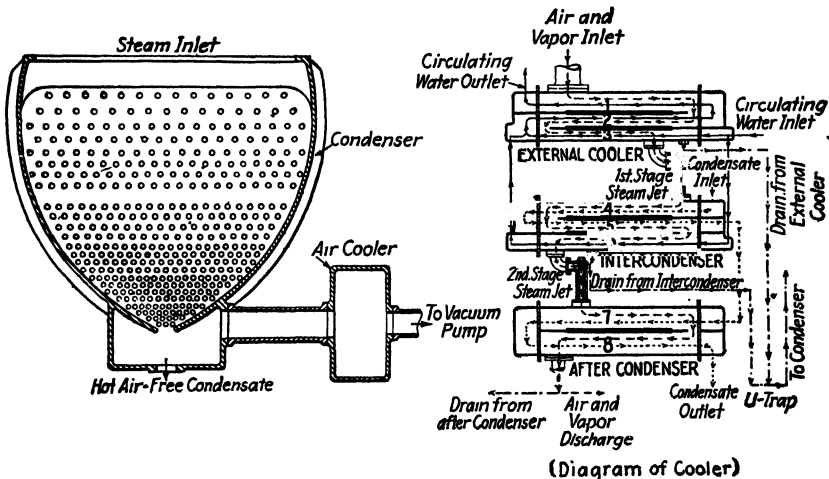



FIG. 263.—Diagrammatic sketch showing the arrangement and operation of a Rand surface condenser equipped with an air cooler.

trend are the limited space available, improved steam economy of turbines and the increased use of the regenerative cycle.

Nearly all of the surface condensers in use in power and industrial plants are built with shell and heads of cast iron and tubes and tube sheets of a special copper alloy. In marine practice steel-plate construction is often used in place of cast iron, and, at the present time, there are indications that welded steel-plate condensers may meet with approval in general engineering practice.

Condenser tubes are available in sizes ranging from ⅝ to 1¼ in. in diameter, composed of one of the following combinations:

1. Copper (pure).
2. Copper, 70 per cent, zinc 30 per cent (brass).
3. Copper, 67 per cent, zinc 33 per cent (brass).
4. Copper, 70 per cent, zinc 29 per cent, tin 1 per cent (admiralty metal).
5. Copper, 60 per cent, zinc 40 per cent (Muntz metal).

The tubes may be worked hot or drawn cold and afterwards annealed.

The tube sheets are made usually of rolled Muntz metal, and the supports of cast iron. The tubes may be secured in the tube sheets by rolling at one end and packing at the other, or they may be rolled or packed at both ends (Figs. 264 and 265).

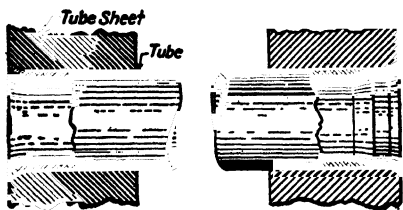



FIG. 264.—Showing condenser tubes rolled into tube sheet.

The packing is usually of fiber, held tight by screwed Muntz-metal ferrules. A shoulder on the inside of the ferrule prevents the tube from creeping out at the end.

Figure 265 shows various types of ferrules used in keeping the tube packing tight.

In case the tubes are rolled at both ends, a floating tube sheet may be used to take care of the tube expansion. In Fig. 266, it is

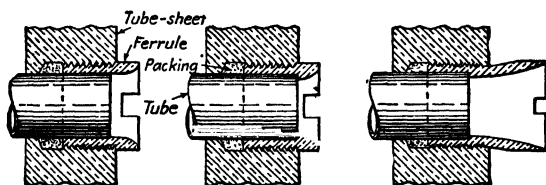



FIG. 265.—Showing method of packing condenser tubes by means of ferrules.

seen that the floating tube sheet may slide on the guide brackets as the tube lengths change. A heavy rubber ring prevents leakage around the tube sheet. Other methods are also used by manufacturers to accomplish the same results.

262. Surface-condenser Maintenance.—*Leakage* of impure cooling water into the steam space is harmful, for the impurities would be carried into the boiler. Leakage is easily determined by measuring the electrical conductivity of the condensate, the conductivity being greater the greater the amount of solids, dissolved or in suspension.

FIG. 266.—Expansion joint for floating tube sheet.

Tube-cleaning methods depend largely on the condition of the tubes as affected by the cooling water used. Loose sediment can be removed by directing a jet of high-pressure water through each tube. Such matter may also be removed from a divided water-box condenser by shutting down one compartment and using all of the cooling water through the other, thus producing a high velocity to clean the tubes.

Scale with organic matter may be removed by emptying the tubes of water and admitting steam to the steam space of the condenser. The resultant baking cracks the scaly material from the tubes.

Soft deposits are removed by shooting soft rubber plugs (Fig. 267 *a*) through the tubes, using air or water pressure. Harder deposits require the use of brushes or scrapers, such as shown in Fig. 267 *b* and *c*. A weak solution of hydrochloric acid is circulated through the tube to remove hard scale but this should be carefully handled.

A

B

C

a. Rubber plug.
b. Scraper-type brush.
c. Wire tube brush.
FIG. 267.—Condenser tube-cleaning accessories.

263. Condenser Auxiliary Equipment.—This includes all equipment, such as pumps, air ejectors, etc., which is necessary for the proper functioning of a condenser. Auxiliary equipment is usually placed adjacent to the condenser which it serves.

Figures 252 and 257 illustrate condensers with the auxiliaries attached. The usual arrangement, however, especially with large condensers, is that in which each piece of equipment is a more or less separate unit, and is located at any convenient point near the condenser.

264. Circulating-water Pumps.—Cooling-water circulating pumps and condensate pumps are generally of the centrifugal impeller type, though reciprocating pumps are sometimes used in small-size units. For the removal of non-condensable gases, steam ejectors or reciprocating vacuum pumps are used. Large units almost invariably employ ejectors.

In large surface condensers, the cooling water is usually siphoned to and from the heads. Pumps are used, in such cases, to promote rapid flow and overcome the friction of the system.

Where the siphon system is not used, the head to be overcome consists mainly of the static head and the friction head. The static head is more or less constant, but the friction head depends on the number of passes, the length and size of tubes and the size and type of the pipe and connections. The velocity head is usually small and is often neglected. The total head may vary from 10 to 50 ft. for surface condensers, and as high as 75 or 80 ft. for jet condensers. Circulating pumps usually range in size of inlet from 10 to 54 in., and

in capacity, at a 20-ft. head, from 2,500 to 90,000 gal. per minute. In common practice centrifugal circulating pumps are driven by electric motors, although steam turbine drive is sometimes resorted to.

Figure 268 shows a photograph of a large circulating pump constructed with a volute type suction chamber and bronze impellers.

In a centrifugal circulating-water pump, the capacity varies directly as the speed; the head, as the square of the speed; and the horsepower

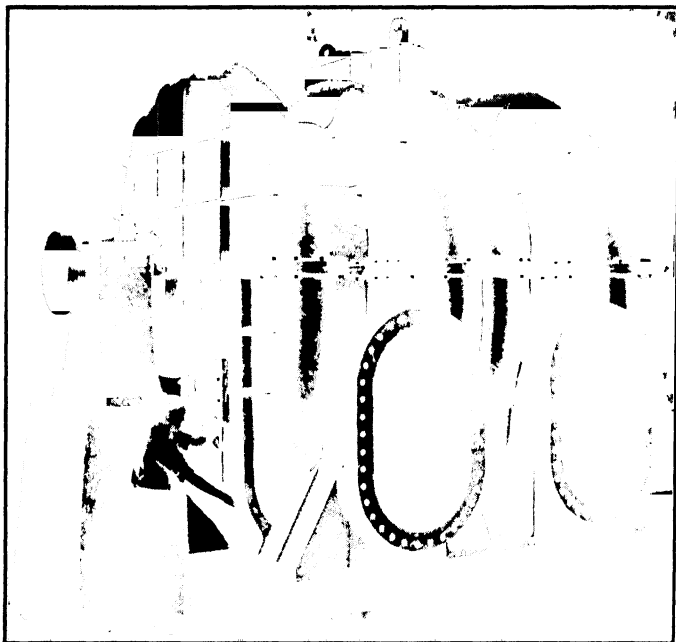



FIG. 268.—Westinghouse single-stage, two-impeller circulating-water pump.

as the cube of the speed. Figure 269 shows characteristic curves for a circulating pump, which bear out these facts.

265. Condensate Pumps.—The condensate is removed from a surface condenser usually by a centrifugal pump called a condensate or hot-well pump. It may be either single or multiple stage. The multiple-stage pumps are used for high-head conditions, and they are operated at constant speed and constant head. Figure 270 shows the construction of a typical two-stage condensate pump.

266. Vacuum Pumping Equipment.—The pumps or ejectors required to remove the air from condensers are of various types of construction. In the case of the unit condenser (Fig. 257), the same pump removes both air and condensate. Such a pump is called a *wet-air pump* or a *wet-vacuum pump*. It is usually the case, however,

that a separate pump is used to remove the air and gases, and this pump is known as a *dry-air* or *dry-vacuum pump.*

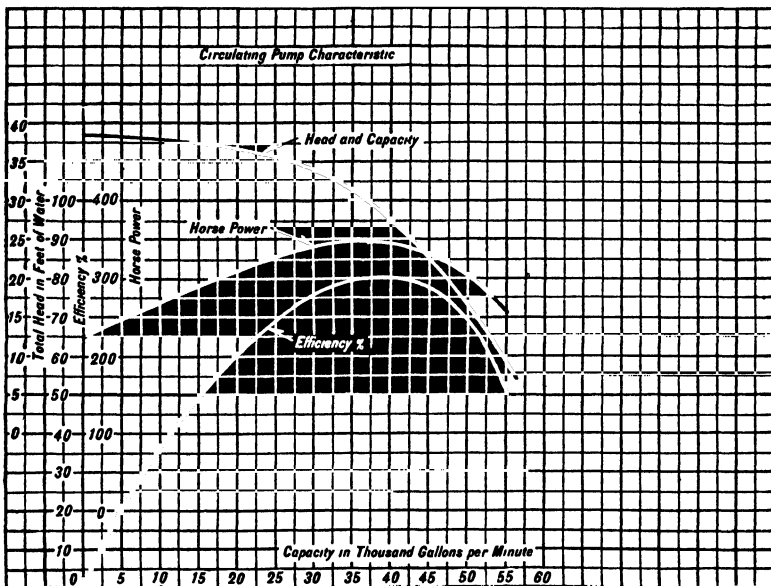



FIG. 269.—Characteristic curves of a circulating-water pump

For removing air, both steam-driven and motor-driven reciprocating pumps are available. The former is used, particularly, in indus-

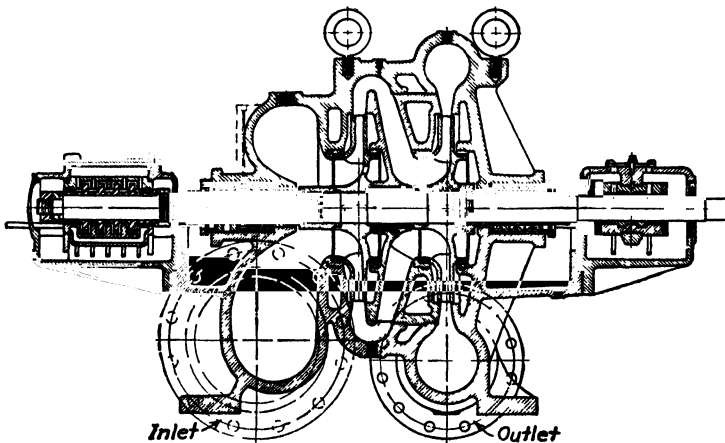



FIG. 270.—Westinghouse two-stage condensate pump.

trial plants, where the exhaust steam may be used for heating buildings. These pumps are either horizontal or vertical, in design, and, if steam driven, the steam and water pistons are arranged on the same piston

rod. This type of pump is used more for wet-vacuum than for dry-vacuum service.

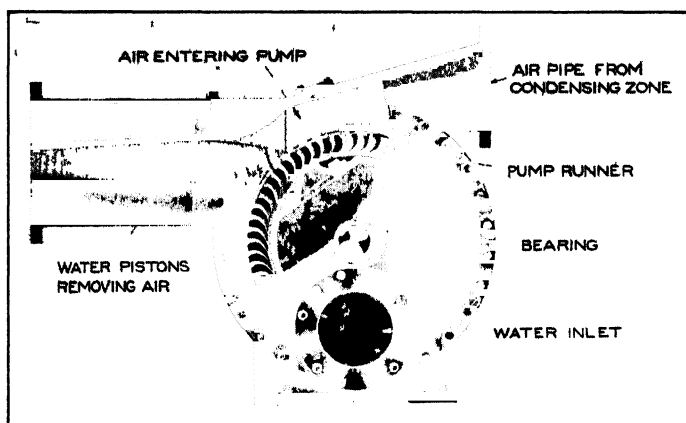



Fig. 271.—Leblanc air pump.

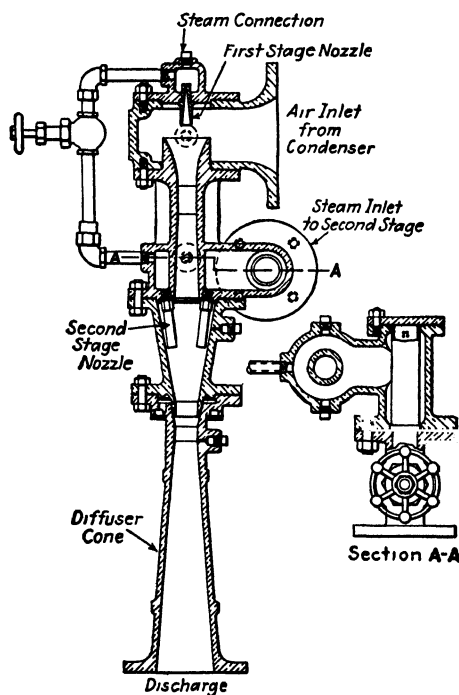



Fig. 272.—Non-condensing, two-stage air ejector (Westinghouse).

Figure 271 illustrates the operation of the Leblanc air pump, which is of the dry-vacuum type. Water from the cooling-water source

is confined to the water chamber, in which the only outlet is through the blades of the rotating runner. These blades discharge the water, in the form of thin pistons, into a collector cone. The non-condensable gases from the condenser are entrained between the series of continuous water pistons, and discharged, with the water, to the atmosphere. This action creates a vacuum in the air pipe which forms a conduit for a continuous supply of air and other gases on flowing from the condenser.

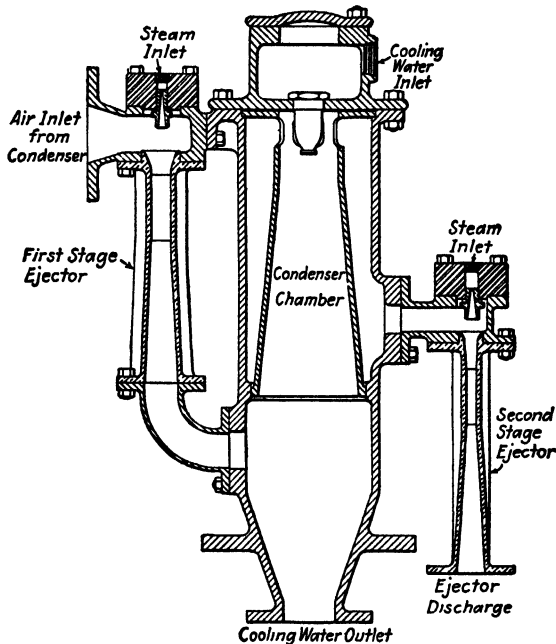



FIG. 273.—Condensing-type air ejector, with intermediate jet condenser (Westinghouse).

The *steam air ejector* removes air by the friction of steam jets. In Fig. 272, illustrating a non-condensing air ejector, it may be seen that the first stage has one steam nozzle, and that the second stage contains multiple steam nozzles. The nozzles discharge into a combining cone, and then into a diffuser cone where the energy due to velocity is transformed into pressure, or potential energy. In this ejector, the second stage handles not only the air but also the steam from the first stage.

Figure 273 shows a cross-section view of a two-stage ejector with a jet condenser between the first and second stage. The purpose of the condenser is to remove the steam consumed in the first stage, thus allowing the second stage to handle air only.

A similar type of ejector using a surface condenser, which is divided into two parts, is shown in Fig. 274. The upper part of the condenser serves as the intermediate and the lower part as the aftercondenser. The condensate from the condenser may be returned to the main-condenser hot well. The cooling water required is obtained from the main circulating-water supply.

In Fig. 275 is illustrated, in section, a two-stage, condensing, multi-steam-jet, vacuum pump. Steam from the first stage ejector nozzles carry the vapor and gases. The steam jets, at a velocity of approximately 2,800 ft. per second, enter the throat of the Venturi tube and create a powerful suction, thus entraining a large volume of

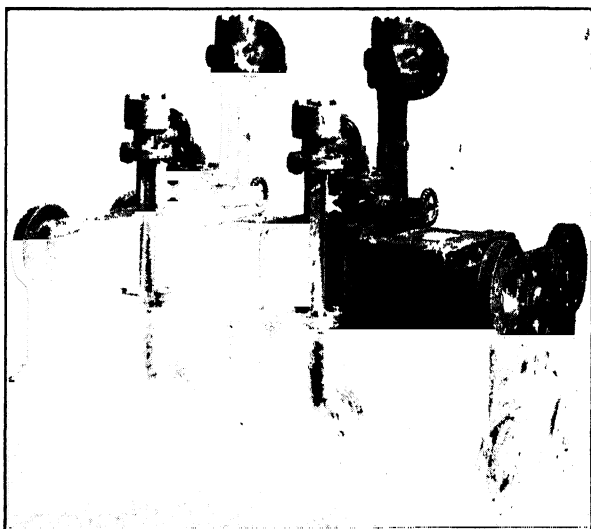



FIG. 274.—Condensing-type air ejector, with intermediate and aftercondenser.

vapor and gas. The discharge enters the intercondenser where the steam is condensed and the gases cooled, reducing them in volume. The non-condensable gases are then drawn by suction into the second-stage ejector pump, where they undergo the same process, being discharged into the aftercondenser at a pressure slightly above that of the atmosphere. The devaporized gases are discharged to the atmosphere, and the condensate drained off.

The multi-stage, steam-ejector vacuum pump is used where high vacuums are required and large volumes of gases must be handled. The capacity varies with the steam pressure and the size and number of steam nozzles. Curves giving comparative performance of ejector pumps are shown in Fig. 276. It may be noticed that the better vacuum is attained by the two-stage pump, even at high capacity.

267. The Radojet Vacuum Pump.—The "Radojet" (Fig. 277) is a special type of steam-operated air ejector. Steam enters the chamber at the left and is divided, a part flowing to the second-stage nozzle

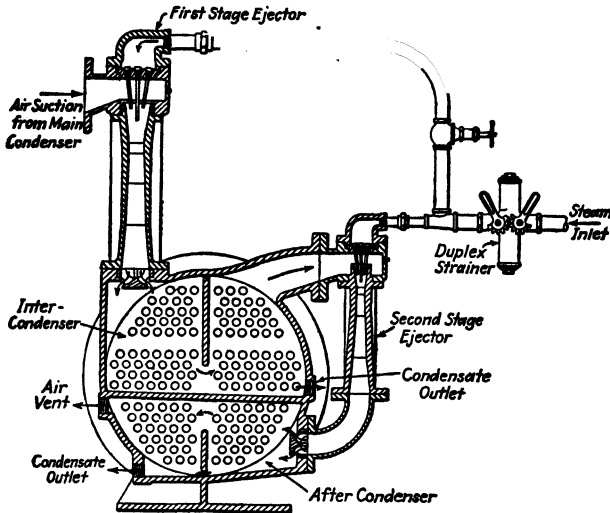



Fig. 275.—Sectional view of a Schutte and Koerting two-stage, steam-jet vacuum pump, with surface inter- and aftercondensers (Vacuator).

and a part passing upward through the small hand valve into the upper nozzle chamber. The steam expands in the first-stage nozzles, acquiring a very high velocity. The jets flowing across the suction chamber entrain air and vapors and pass into a diffuser. Here the

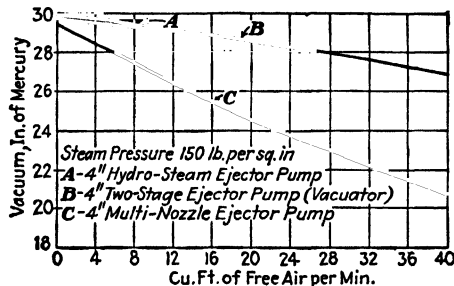



Fig. 276.—Curves showing capacity of air-removal equipment.

velocity is changed into pressure and the mixture discharged into the second-stage suction chamber at a higher pressure. The steam used by the second stage expands radially through an annular nozzle. The nozzle point is adjustable and can be moved toward or away from the nozzle, thus changing the cross-section of the passage through

which the steam flows. The steam, at high velocity, leaves the second-stage nozzle in the form of a thin sheet and entrains the mixture of air and steam from the suction chamber. By this means it is compressed and discharged to the atmosphere.

The Radojet is also used with an intermediate surface condenser placed between the two stages. This reduces the steam consumption by approximately one-half. In cases of high vacuum, a three-stage Radojet is used. In these installations intermediate condensers are used between each stage and an aftercondenser following the last stage.

In the vacuum pump illustrated in

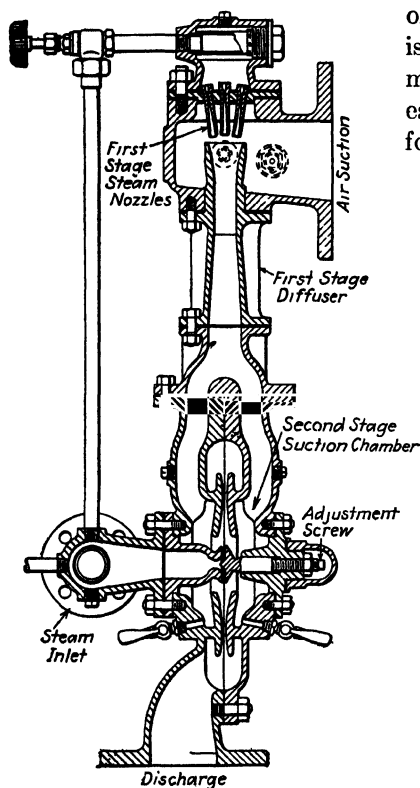



FIG. 277.—Radojet vacuum pump.

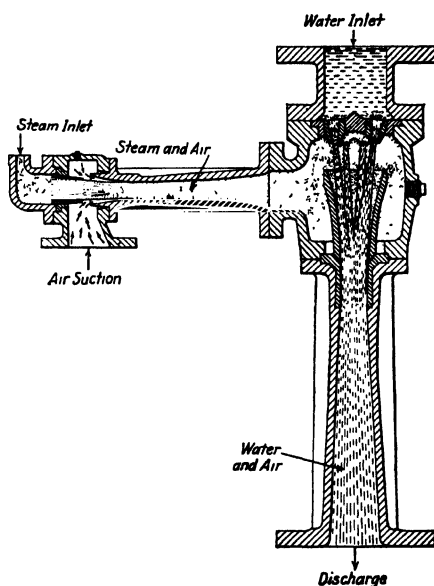



FIG. 278.—Multi-nozzle, hydro-steam vacuum pump (Schutte and Koerting.)

Fig. 278, the first stage consists of a steam-jet ejector and the second stage a water-jet ejector which acts, also, as a jet condenser for the vapors leaving the first stage. Curve A (Fig. 276) shows the capacity range for this type of pump, with steam at 150 and water at 15 lb. per square inch pressure.

268. Heat Absorbed per Pound of Exhaust Steam.—The heat absorbed from 1 lb. of exhaust steam in a condenser may be determined by finding the enthalpy in 1 lb. of steam at the turbine throttle, and

subtracting from this the heat equivalent of the work of the turbine and the heat loss due to radiation. As the output of the turbine is usually measured at the generator, no accurate determination of the losses in the unit can be made. The mechanical efficiency of the unit may be assumed from general practice.

The following example illustrates a method for calculation.

Example 13-1.—A turbine is supplied with steam at a pressure of 225 lb. per square inch absolute and a temperature of 550°F.; exhaust pressure 1.6 in. of mercury absolute. The water rate is 12 lb. per kilowatt-hour, with a mechanical efficiency of 96 per cent. Assume 1 per cent radiation and friction loss for the turbine, and calculate the heat absorbed by the circulating water from each pound of exhaust steam.

Solution.—Based on 1 lb. exhaust steam:

Enthalpy of initial steam	=	1,293.0 B.t.u.
Heat equivalent of the work, $\frac{3{,}414}{12.0 \times 0.96 \times 0.99}$	=	299.7
Enthalpy of exhaust steam	=	993.3
Heat rejected in condensate	=	61.8
Heat absorbed by cooling water	=	931.5 B.t.u.

A method for determining the heat absorbed by the condenser per pound of exhaust steam is to assume constant-entropy expansion in the turbine and, from common practice, the friction loss through the turbine. The friction loss results chiefly in a reheating of the steam. Using the conditions of Example 13-1, and assuming a friction loss of 15 per cent, the solution is as follows:

Solution:

Entropy of initial steam	=	1.6359
Enthalpy initial steam	=	1,293.0 B.t.u.
Enthalpy exhaust steam at constant entropy	=	902.0
Heat drop	=	391.0
15 per cent friction reheating	=	58.7
		902.0
Corrected exhaust enthalpy	=	960.7
Heat rejected in condensate	=	61.8
Heat absorbed		898.9 B.t.u.

Both methods are approximations and commercial calculations are customarily based on a heat absorption of 950 B.t.u. per pound

of exhaust steam, at the usual absolute condenser pressure of 1 to 1½ in. of mercury.

269. Injection Water Required in Jet Condensers.—The weight of injection water required for a jet condenser may be determined as follows:

$$W(h_{f2} - h_{f1}) = h_3 - h_{f2}$$

or

$$W = \frac{h_3 - h_{f2}}{h_{f2} - h_{f1}}$$

or

$$= \frac{970}{t_2 - t_1} \qquad (151)$$

in which

W = weight of cooling water per pound of exhaust steam, lb.
h_{f1} = enthalpy per pound of cooling water entering, B.t.u.
h_{f2} = enthalpy per pound of hot water leaving, B.t.u.
h_3 = enthalpy in exhaust steam, B.t.u. per pound.
t_2 = temperature of mixture of condensate and injection water leaving condensate, °F.
t_1 = temperature of entering injection water, °F.

270. Cooling Water Required in Surface Condensers.—For surface condensers, the weight of cooling or circulating water required per pound of exhaust steam may be determined by the following equation:

$$W(h_{f2} - h_{f1}) = h_3 - h_{f4}$$

$$W = \frac{h_3 - h_{f4}}{h_{f2} - h_{f1}}$$

or

$$= \frac{950}{t_2 - t_1} \qquad (152)$$

in which

W = weight of circulating water per pound of exhaust steam, lb.
h_{f4} = enthalpy per pound of condensate, B.t.u.
t_1 and t_2 = temperature of circulating water at inlet and discharge, respectively, °F.
t_3 = temperature of exhaust steam, °F.
t_4 = temperature of condensate, °F.

Other items as in Eq. (151).

271. Cooling Surface in Surface Condensers.—The *cooling surface* of a surface condenser is determined from the unit of conductivity of the tube walls, the heat to be transmitted, and the mean temperature difference between the steam and the circulating water. In the condenser chamber, the temperature of the steam and condensate is approximately constant but the temperature of the water in the tubes increases as the heat is absorbed. The final temperature of the water gradually approaches the temperature of the steam, as it flows through the condenser, which causes the heat flow through the tube walls to decrease. There is, therefore, an exponential relation between the heat flow and the mean temperature difference between the steam and cooling water. The equation of heat flow is as follows:

$$q = UAD_m \tag{153}$$

in which

q = heat transmitted, B.t.u. per hour.
U = heat absorbed, B.t.u. per hour, per square foot of surface per degree Fahrenheit. [U varies from about 350 to a maximum of 800 to 900. An average value would be 400 for a water velocity in the tubes of 1 ft. per second to 600 for a velocity of 5 ft. per second.]
A = total condenser surface, sq. ft.
D_m = logarithmic mean temperature difference, °F.

$$D_m = (t_2 - t_1) \div \left(\log_e \frac{t_3 - t_1}{t_3 - t_2}\right) \tag{154}$$

in which the temperatures are the same as used in Eq. (152).

For approximate calculation, instead of the logarithmic mean temperature difference as given above, the arithmetical mean temperature difference may be used. The latter is as follows:

$$D_m = t_3 - \left(\frac{t_2 + t_1}{2}\right)$$

Example 13-2.—From a test on a 35,000-kw. turbogenerator, the following data were taken: steam rate 11.85 lb. per kilowatt-hour; pressure at throttle 350 lb. per square inch absolute; steam temperature 675°F.; exhaust pressure 2 in. of mercury absolute. Heat lost by radiation is assumed to be 1 per cent of the initial heat in the steam. The condenser data: cooling-water temperatures, inlet 65°F., discharge 81°F.; temperature of condensate 95°F.; water velocity in tubes 12 ft. per second; unit of heat conductivity, 650 B.t.u. per hour per square foot of surface per degree temperature difference between steam and water; outside diameter of tubes, 1 in.; thickness of tube walls, 0.046 in. Determine (*a*) ratio

of condenser surface to the kilowatt capacity of the turbine, (*b*) number of tubes required for a two-pass surface condenser, (*c*) length of tubes.

Solution:

Enthalpy per pound of steam at throttle, h	=	1,351.2 B.t.u.
Work done per pound of steam, 3,414 ÷ 11.85	=	288
Difference	=	1,063.2
Radiation loss, 0.01 × 1351.2	=	13.5
Enthalpy per pound of exhaust steam, h_3	=	1,049.7
Enthalpy per pound of condensate, h_{4f}	=	62.96
Heat per pound of steam rejected in condenser	=	986.74
Cooling water per pound of steam, 986.74 ÷ 16	=	61.6 lb.
Cooling water per hour, 61.6 × 11.85 × 35,000	=	25,550,000 lb.

The mean temperature difference is by Eq. (154),

$$D_m = (81 - 65) \div \log_e \left(\frac{101.17 - 65}{101.17 - 81}\right) = 27.5$$

From Eq. (153), the surface area is

$$A = (25{,}550{,}000 \times 16) \div (650 \times 27.5) = 22{,}850 \text{ sq. ft.}$$

a. Ratio tube surface to kilowatts, 22,850 ÷ 35,000 = 0.654

Water per second (25,550,000 × 0.01604) ÷ 3,600 = 114 cu. ft.

Water per second per tube ($\pi \times 0.908^2 \times 12$) ÷ (4 × 144) = 0.054 cu. ft.

b. Number of tubes required (2 × 114) ÷ 0.054 = 4,230

Surface area per tube 22,850 ÷ 4,230 = 5.41 sq. ft.

c. Length of tube (5.41 × 12) ÷ π × 1 = 20.6 ft.

272. Cooling-water Supply.—For the condensation of 1 lb. of exhaust steam, from 50 to 70 lb. of cooling water are required. As turbine units of 100,000-kw. capacity become more common, using approximately 1,000,000 lb. of steam per hour, it may be seen that the large central stations need an enormous supply of cooling water for the condensers. It is said that the Lake Shore station of the Cleveland Electric Illuminating Company uses more water for the condensers than the entire city of Cleveland uses in the mains. Whenever possible, large power stations are located so as to have available an unlimited supply of cooling water, usually from a river or lake. A station not so situated has to make provision for using, repeatedly, the same cooling water. It would be far too expensive to use city water for condensers.

The method commonly used in overcoming a condition of limited cooling-water supply is to repeatedly cool and recirculate it through

the condenser. Means for accomplishing the cooling process are as follows:

1. Cooling pond.
2. Spray pond.
3. Cooling tower.

In all cases, the cooling is effected by exposing a large surface of water to currents of atmospheric air. This evaporates a part of the water and the heat required for evaporation is absorbed from it and carried away by the air. The degree of cooling depends on the air and water temperatures, the humidity of the air, and the quantity of air brought into contact with the water.

273. The Cooling Pond.—This, of the above systems, is the simplest and consists of cooling the water in open ponds. This method is ineffective unless a very large pond is provided and, for this reason, it is used with small condensers only.

274. The Spray Pond.—In the spray-pond system, the hot cooling water from the condensers is discharged over a pond from special spray nozzles. The cooling occurs while the water, in the form of drops and mist, is in contact with the air. A small part is vaporized and the remainder drops in the pond or reservoir. There are different

TABLE 13-1.—WATER RATIOS WITH COOLING EFFECTS, SPRAY PONDS[1]

Air temperature, °F.	Temperature, cooling water		Vacuum, in. Hg.	Water ratio
	Entering condenser, °F.	Leaving condenser, °F.		
60	69 1	93.1	28 15	40:1
90	98.6	122.6	25.8	40:1
60	66.6	85.5	28.55	55:1
90	95.5	114.5	26.6	55:1
60	65.0	81.0	28.75	60:1
90	93.0	109.0	27.05	60:1
60	64.0	78.0	28.85	70:1
90	89.2	103.2	27.48	70:1

[1] Binks Spray Equipment Company.

types of nozzles used for effecting the spray. Some are designed to discharge the water in the form of cones while others deliver a fan-shaped spray. The nozzles are provided in sufficient number, and the arrangement is such that the sprays do not interfere with each other. The spray pond usually occupies considerable area and is placed as close as possible to the plant.

The water loss due to evaporation is generally estimated at about 2 per cent. Roughly, the same weight of water is evaporated in the spray pond as is condensed in the condenser. In addition, there is some driftage loss, due to the spray being blown from the pond. This can not be estimated with any accuracy, but the total loss of water may be assumed to be 5 per cent under conditions where high driftage occurs.

Table 13-1 gives the approximate temperatures encountered in the usual spray pond, with different ratios of cooling water to steam condensed, based on a relative humidity of 70 per cent.

275. Cooling Towers.—Cooling towers occupy a smaller area than spray ponds and eliminate the water spray which scatters, to a certain extent, over the surrounding ground, and which, especially in winter, is undesirable. Cooling towers are often placed on the roof of the power plant, and, in some industrial plants, the cooling tower has a shape to resemble the product of the plant. For instance, a large dairy plant has the cooling tower disguised as a large milk bottle.

Cooling towers are classified as (1) *atmospheric,* (2) *natural draft* or *chimney,* and (3) *forced draft.* The principle of cooling is the same for all classes of towers. The hot water is pumped to the top and falls in a spray or in thin sheets over trays, screens or other filling until it reaches the reservoir at the bottom. Air enters at the bottom and passes upward, through the water, and leaves at the top. The air absorbs heat from the water by radiation and convection.

In the *atmospheric cooling tower,* the water is sprayed onto the top distributing deck, where the deck sections have grooves or gutters which distribute the water evenly over the whole area of the tower. It then flows down between deck sections to the splash deck immediately below. This is repeated until the reservoir is reached. Louvres are arranged in the side walls to permit the air to flow in freely. For winter operation, a secondary set of spray nozzles is often provided near the bottom of the tower. This reduces the cooling effect and prevents freezing. All sides, in this type of tower, are open to the atmosphere.

The *natural-draft* or *chimney* type of tower has openings at the bottom through which the air enters, the cooling chamber is completely encased, and above the cooling chamber is an enclosed tower which has the same function as a chimney. The warm humid air is drawn up and discharged at the top, thus inducing a continual flow of air through the tower.

The *forced-draft tower* is also completely enclosed, with discharge openings at the top and fans at the bottom to produce the flow of air.

Figure 279 shows the construction of a cooling tower of this type. The water, from a series of nozzles at the top, falls over a staggered arrangement of triangular-shaped filling pieces, splashing from one piece to another until it reaches the reservoir at the bottom. Air is blown

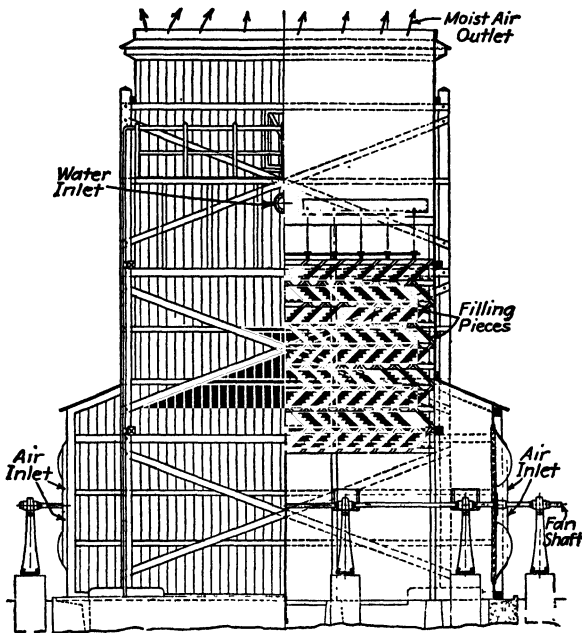



FIG. 279.—Forced-draft cooling tower.

in by two fans, one at each side of the tower, passes upward and is discharged at the top.

Problems

1. A two-pass surface condenser condenses 30,000 lb. steam per hour; cooling water enters at 56°F. and leaves at 80°F.; the vacuum in the condenser is 27.8 in. of mercury; barometer 29.8 in. of mercury; temperature of condensate 90°F.; quality of exhaust steam 89.8 per cent. In the tubes, the water velocity is 8.5 ft. per second; outside diameter of tubes 1 in.; thickness 0.049; $U = 750$. Determine (*a*) area of tube surface required, (*b*) number of tubes, (*c*) length of tubes.

2. A jet condenser maintains a vacuum of 25.4 in. of mercury; barometer 28.95 in. of mercury; cooling water temp. 65°F.; discharge temperature 82°F. The steam engine served by the condenser uses 25 lb. steam per brake horsepower-hour; mechanical efficiency 89 per cent; initial steam pressure 140 lb. per square inch absolute; steam temperature 360°F. Determine the weight of water required per pound of steam.

3. A steam engine has a water rate of 21.2 lb. steam per indicated horsepower-hour; initial steam pressure 155 lb. per square inch absolute, quality 98.7 per cent, exhausts into a jet condenser of a vacuum of 22.8 in. of mercury; barometer

29.55 in. of mercury. How much cooling water will be required per pound of steam if the inlet temperature is 58°F., and the discharge temperature is 78°F.?

4. A steam turbine uses 15 lb. of steam per kilowatt-hour; steam pressure 200 lb. per square inch absolute; 110° of superheat; condenser vacuum 28.5 in. of mercury; barometer 30 in. of mercury. In the jet condenser serving the turbine, the cooling water enters at 55°F., and discharges at 82°F. Assuming that 1 per cent of the initial enthalpy is lost by radiation, determine the weight of water required per pound of steam.

5. A steam engine uses 17.5 lb. steam per brake horsepower-hour. The mechanical efficiency of the engine is 85 per cent; initial steam pressure 154 lb. per square inch absolute; 55° of superheat; vacuum 23.8 in. of mercury; barometer 28.8 in. of mercury. The surface condenser used with the engine receives cooling water at a temperature of 68°F., and discharges it at 100°F.; temperature of condensate, 115°F. Heat lost by radiation is 1 per cent of the initial enthalpy of the steam supplied to the engine. Determine the weight of cooling water required per pound of steam.

6. A turbine, with a surface condenser, uses 12.5 lb. steam per kilowatt-hour; initial steam pressure 200 lb. per square inch absolute; temperature 580°F.; exhaust at 27.9 in. of mercury vacuum; barometer 29.4 in. of mercury. The radiation loss is 1 per cent of the available heat energy. The condenser receives cooling water at 70°F.; discharges at 85°F. Determine the weight of cooling water required per pound of steam.

7. A 5,000-kw. turbine has a water rate of 12.5 lb. per kilowatt-hour when operating on steam at 164 lb. per square inch absolute pressure and, 74.5° of superheat; condenser vacuum, 26.5 in. of mercury; barometer 29.5 in. of mercury. The radiation loss is assumed as 1 per cent of the initial enthalpy of the steam. Cooling-water temperatures: inlet 65°F.; discharge 82°F. Determine the square feet of tube surface required in the surface condenser for this unit, using value 400 for U. (Condensate leaves the condenser at 5° below the saturation temperature.)

8. A 35,000-kw. turbine uses 11.5 lb. of steam per kilowatt-hour. Initial steam pressure 325.5 lb. per square inch gage; superheat 200°; vacuum 28 in. of mercury; barometer 29.5 in. of mercury. Radiation loss 1 per cent of the initial enthalpy. Cooling-water temperatures, entering 65°F.; discharge 82°F. Condensate temperature 89°F.; $U = 700$; outside diameter of tubes is 1 in.; thickness 0.049 in.; water velocity in tubes 10 ft. per second. Determine (*a*) weight of cooling water required per hour, (*b*) the area required for a surface condenser, (*c*) the number of tubes, and (*d*) the length of tube required (single-pass condenser).

9. A 52,500-kw. turbine receives steam at 600 lb. per square inch absolute pressure, and 725°F. At 130 lb. per square inch absolute, the steam is withdrawn from the turbine and reheated, at constant pressure, to 720°F.; exhaust pressure at 1 in. of mercury absolute. The turbine nozzle efficiency is 85 per cent. The heat content of the exhaust steam may be determined from the steam path on the Mollier diagram. Steam flow at throttle, 490,000 lb. per hour, of which 150,000 lb. per hour are extracted at different points through the turbine. A single-pass surface condenser is used; cooling water inlet temperature is 66°F.; discharge temperature 75°F.; condensate temperature 78°F.; $U = 580$; outside tube diameter, 7/8 in.; thickness 0.046 in.; water velocity 8 ft. per second. Determine (*a*) weight of cooling water per hour, (*b*) ratio of condensing surface to kilowatt capacity of turbine, (*c*) number of tubes, and (*d*) length of tube.

CHAPTER XIV

PUMPING EQUIPMENT

276. Introductory.—A pump is a machine used to effect the transportation of a fluid (usually a liquid) from one location to another. As a general rule, a pump has as its function, not only that of transportation, but also the imparting of pressure to the liquid medium. In power plants, pumps are used, principally, for handling water over a wide range of temperatures. Air pumps and other equipment for handling gaseous fluids have been discussed in the previous chapters, and, therefore, will not be taken up here.

277. Classification of Pumps.—Pumps commonly used for handling water in the service of power plants are classified as follows:

1. Reciprocating, piston or plunger pumps.
2. Centrifugal or impeller pumps.
3. Jet pumps.

In addition to those found in the above classification are pumps the operation of which involves any one of a large variety of principles. These, if used, are not of any great importance in the operation of power plants and will not be taken up in this text. Pumps of the above classification will be discussed in the following articles.

Reciprocating pumps may be subclassified as (1) *direct-acting pumps*, (2) *flywheel pumps*, and (3) *power-driven pumps*.

Centrifugal pumps may be subclassified, also, as (1) *volute pumps*, and (2) *turbine pumps*.

278. Direct-acting Pumps.—Direct-acting steam pumps are simple in design, flexible in operation, low in first cost and inexpensive to maintain. The steam piston and water piston are on the same piston rod, and no crank or flywheel is included in the design. Both steam and water cylinders are double acting. The steam valves are operated either by a connection to the piston rod or by the steam cylinder piston. Slide valves without lap or lead are commonly used, which provision admits steam at full pressure into the cylinder for practically the entire length of the stroke. The steam consumed is not used expansively; otherwise the pump would stop when the force of the steam pressure becomes less than the force due to the pressure in the water cylinder.

A *simplex* direct-acting steam pump is a single-cylinder pump; that is, with one steam cylinder and one water cylinder. A *duplex* direct-acting steam pump consists, essentially, of two single-cylinder pumps mounted side by side. Both steam cylinders use high-pressure steam and exhaust to a common exhaust line, and both water cylinders take water from a common supply line and discharge into a common outlet pipe.

Figure 280 shows a longitudinal section of one side of a direct-acting, duplex pump. The operation is such that as one piston completes its stroke, the piston rod, through a rocker arm, actuates the steam valve of the other cylinder. When one steam piston is at the head end, the other is at the cradle end of its cylinder. When in motion, the pistons travel in opposite directions to each other. The

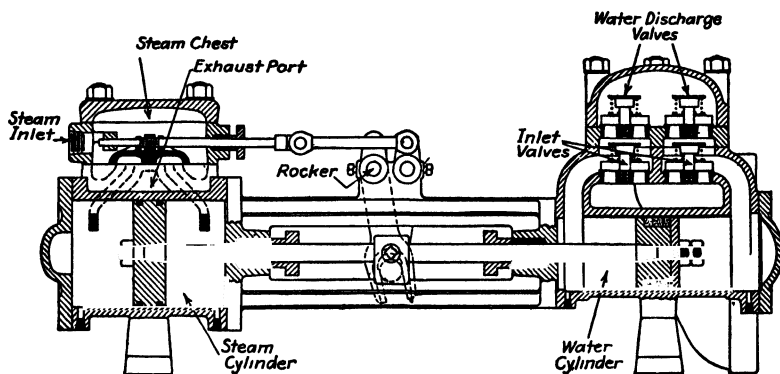



FIG. 280.—Sectional view of the Dean duplex direct-acting pump.

nut on the valve stem that operates the valve is provided with considerable clearance, as shown, so the valve will not be shifted until the piston is near the end of the stroke.

Figure 281 illustrates the relation of the steam valves and pistons at four positions of the stroke. One cylinder is equipped with a D valve and the other with a B valve, as shown in the figure. When the piston is at the end of its stroke, the main port is covered by the piston. A small starting port is provided which is first uncovered by the valve, admitting steam into the clearance space and starting the piston on its return stroke. The valve then opens the main port, increasing the flow of steam as the piston continues on its stroke. At the same time, the other end of the cylinder is open to the exhaust. The large port at each end serves as both a steam and an exhaust port. When the piston covers the exhaust port, the steam trapped in the cylinder acts as a cushion to stop the piston. This diagram shows the relative position of the two pistons during operation.

A steam-actuated valve sometimes used on duplex pumps is illustrated in Fig. 282. Its principle of operation is much different from the steam valve shown in Fig. 280. The rocker arm moves a small slide valve that controls the steam flow to and from the two

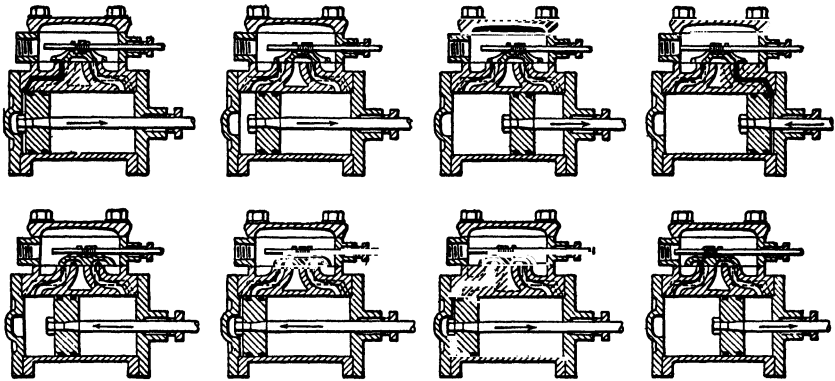



FIG. 281.—Illustrating action of steam valves, Dean duplex pump.

ends of a piston valve. The motion of the piston valve is transmitted to a small D slide valve which covers and uncovers the cylinder ports for proper admission and exhaust of steam.

The water flow at each end of each water cylinder of the pump illustrated in Fig. 280 is controlled by one suction valve and one

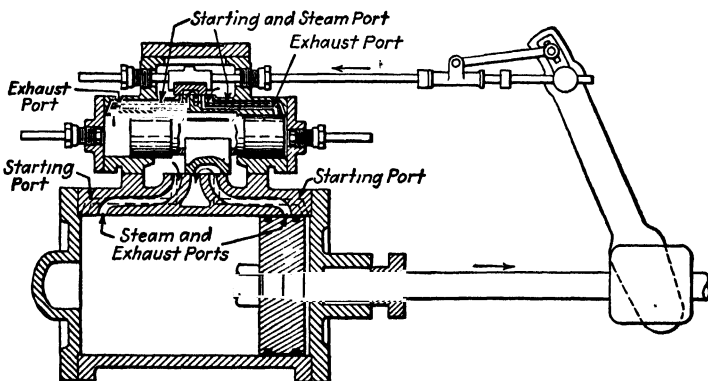



FIG. 282.—Steam-operated valve, Dean pump.

discharge valve. These valves, as shown in Fig. 283, are spring-loaded, flat discs, which also act as check valves, allowing flow in but one direction. For comparatively low-pressure service rubber discs may be used, but for high pressures metallic disc valves (Fig. 283) are required.

Figure 284 illustrates a type of direct-acting pump used for boiler feed service where high water pressures are necessary. Instead of one water piston, two plungers, held together by side rods, operate in a divided cylinder, as shown, which consists of essentially two

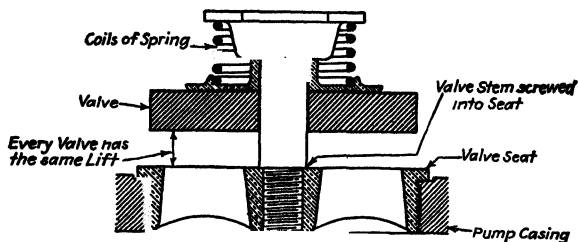



FIG. 283.—Disc water valve used on the Dean duplex pump.

separate cylinders. The steam valve mechanism is similar to that shown in Fig. 282.

The water valves are of the wing-guided type, commonly used with high pressures. With the end-packed type of plunger, the stuffing boxes are visible, and leakage can be readily detected and stopped by tightening the packing glands, while the pump is in operation.

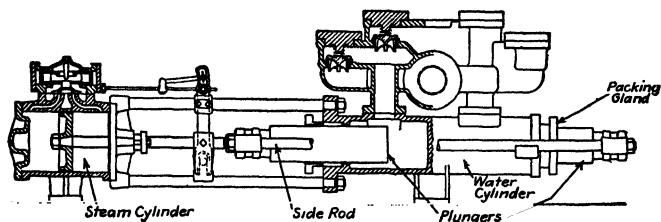



FIG. 284.—Dean pressure pump, outside end packed.

The double plunger and outside packing features have been incorporated in the design of crank-driven, boiler feed pumps for pressures as high as 1,500 lb. per square inch.

A sectional view of the valve chest and steam cylinder of a typical single-cylinder, direct-acting, steam pump is shown in Fig. 285. The valve motion is effected by the main piston. High-pressure steam enters at the top of the valve chest and its flow to the cylinder is controlled by a floating piston valve. A small upper passage admits steam to both ends of the floating valve, and, with the same pressure acting on each end, the valve becomes balanced and remains in a stationary position. When the piston reaches the end of the stroke it strikes and opens the reverse valve. Opening this valve exhausts the steam from one end of the piston valve at a greater rate than that

at which it is admitted, which immediately shifts the floating valve and reverses the motion of the pump piston. As soon as the piston leaves the reverse valve, the steam pressure back of the valve closes it automatically. At each end of the valve chest is a small tappet rod which may be used to reverse the piston valve if for any reason it should become lodged.

279. Flywheel and Power-driven Pumps.—Pumps in these two classes may have either single- or double-acting water cylinders.

Flywheel pumps are known as pumping engines. They use steam expansively, as sufficient energy can be stored in the flywheel to maintain the speed during that part of the stroke in which reduced

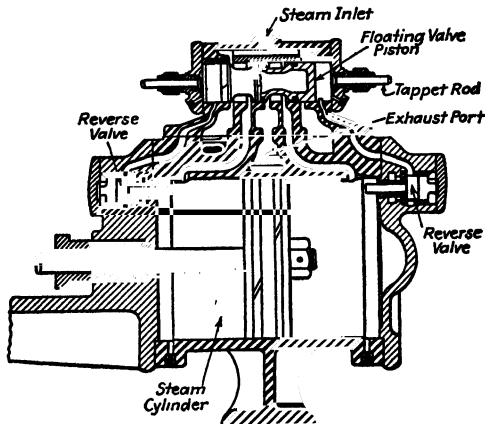



FIG. 285.—Steam cylinder of a Cameron single, direct-acting pump.

steam pressure occurs. Pumping engines run at higher speeds than direct-acting pumps and, consequently, give better economy. They are, however, higher in first cost and usually are more expensive to maintain.

Power-driven pumps are those driven by an external machine, that is, steam engine, internal-combustion engine, or electric motor, with the latter drive predominating. This type of pump is a constant-speed machine, and a spring relief valve is provided in the discharge line to prevent overloading the motor or damaging the pump. Both of these classes of pumps are built as simplex, duplex, or triplex pumps.

280. Centrifugal Pumps.—The centrifugal pump consists of one or more rotors, termed *impellers*, revolving in a fixed plane, within a stationary air-tight casing. Electric-motor or steam-turbine drive is generally used with pumps of this type.

In each type of centrifugal pump, the water enters the rotating impeller at the center and is thrown outward by centrifugal force.

The water, leaving the impeller, has a maximum kinetic energy due to its velocity, and the casing is designed to change, as much as possible, this kinetic energy into potential energy or pressure.

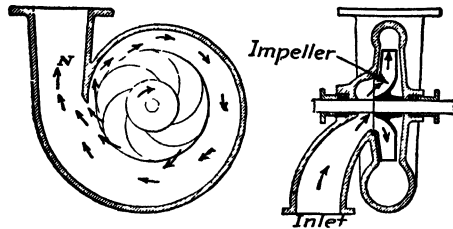



Fig. 286.—Volute type of centrifugal pump.

In *volute pumps* (Fig. 286) the casing forms a discharge chamber of gradually increasing size around the impeller. In this spiral chamber, velocity head is converted into discharge pressure. *Turbine*

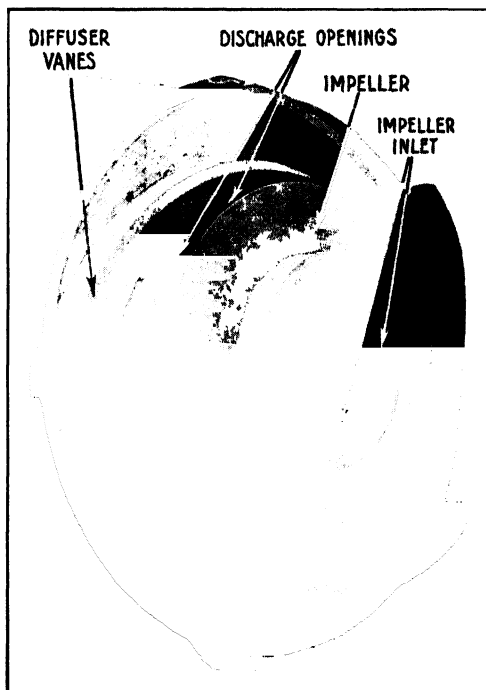



Fig. 287.—Closed-type impeller with diffuser vanes.

pumps have diffusion vanes around the periphery of the impeller, and these vanes have water passages with gradually increasing areas, toward the outside diameter. This arrangement effects the conversion of velocity into pressure the same as the volute discharge chamber.

Centrifugal pumps may be *single stage* or *multi-stage,* depending on the discharge pressure required. The multi-stage pump is commonly of the turbine type. A *single-suction* pump is one in which water enters the impeller from one side, and a *double-suction* pump is one in which water enters both sides. Impellers are classed as of either the open or closed type. The *open impeller* consists of a circular disc with blades on each side. The *closed impeller* has blades similar to the open type but with circular discs shrouding the sides of the blades. Fig-

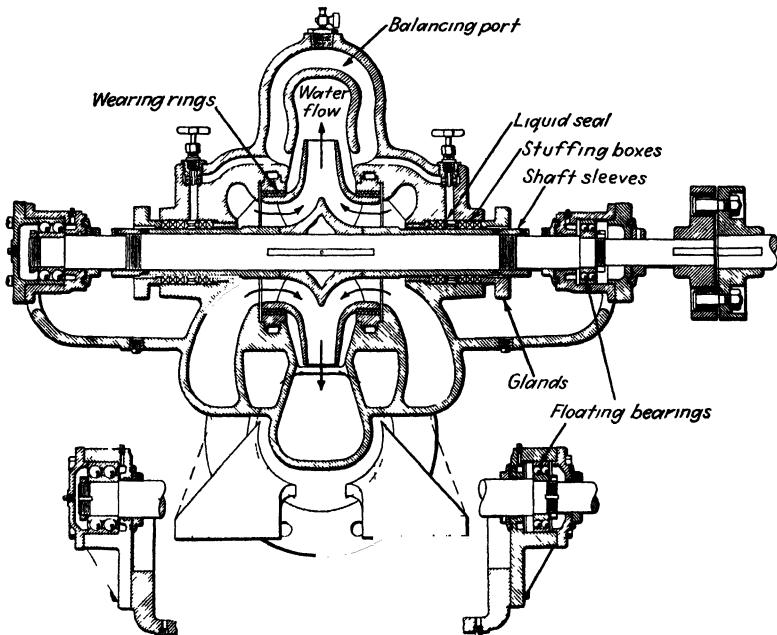



FIG. 288.—Cross-sectional view to illustrate construction of Cameron ALV and AFC double-suction, volute pumps.

ure 287 shows a closed impeller with diffuser vanes. The water enters the impeller through the annular space at the center and is discharged through the diffuser vanes.

Figure 288 shows a sectional view of a single-stage, double-suction, volute pump with closed impeller. For high speeds a Kingsbury thrust bearing (Fig. 289) is provided. For low heads and speeds the thrust bearing is not included in the design, as the double suction results in a hydraulic balance on the impeller.

A two-stage, volute pump used in delivering water from a vacuum, as from a condenser hot well, to a pressure above that of the atmosphere is illustrated in Fig. 290. Closed impellers with single suction are used in this pump. The impellers are opposed to each other so as

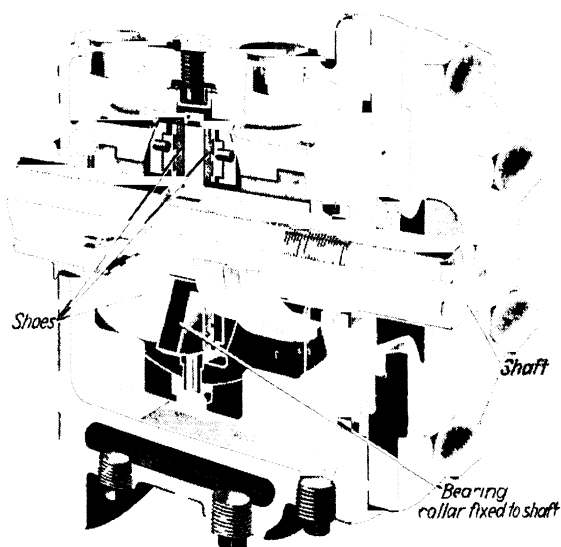



FIG. 289.—Sectional wash drawing of Kingsbury thrust bearing with suction-ring lubrication as used on Cameron pumps that have a double-extended shaft.

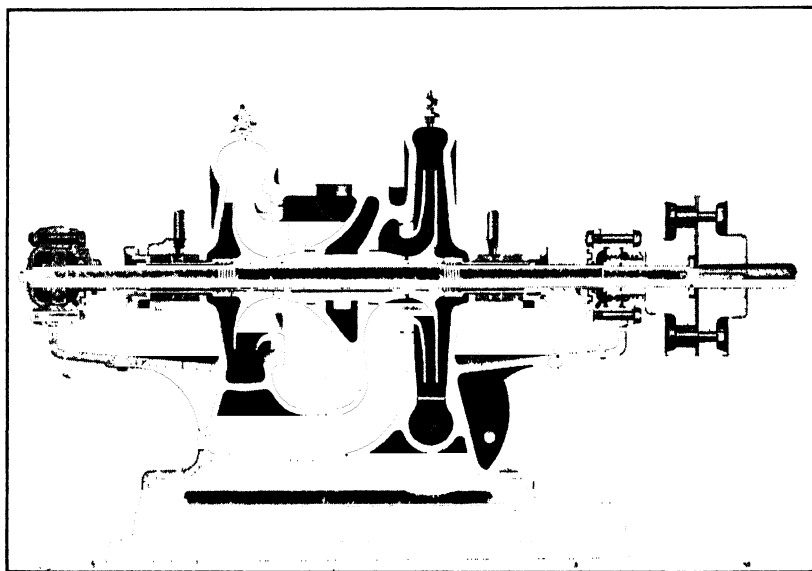



FIG. 290.—Cameron high-head condensate pump.

(a) Longitudinal section.

(b) End view.

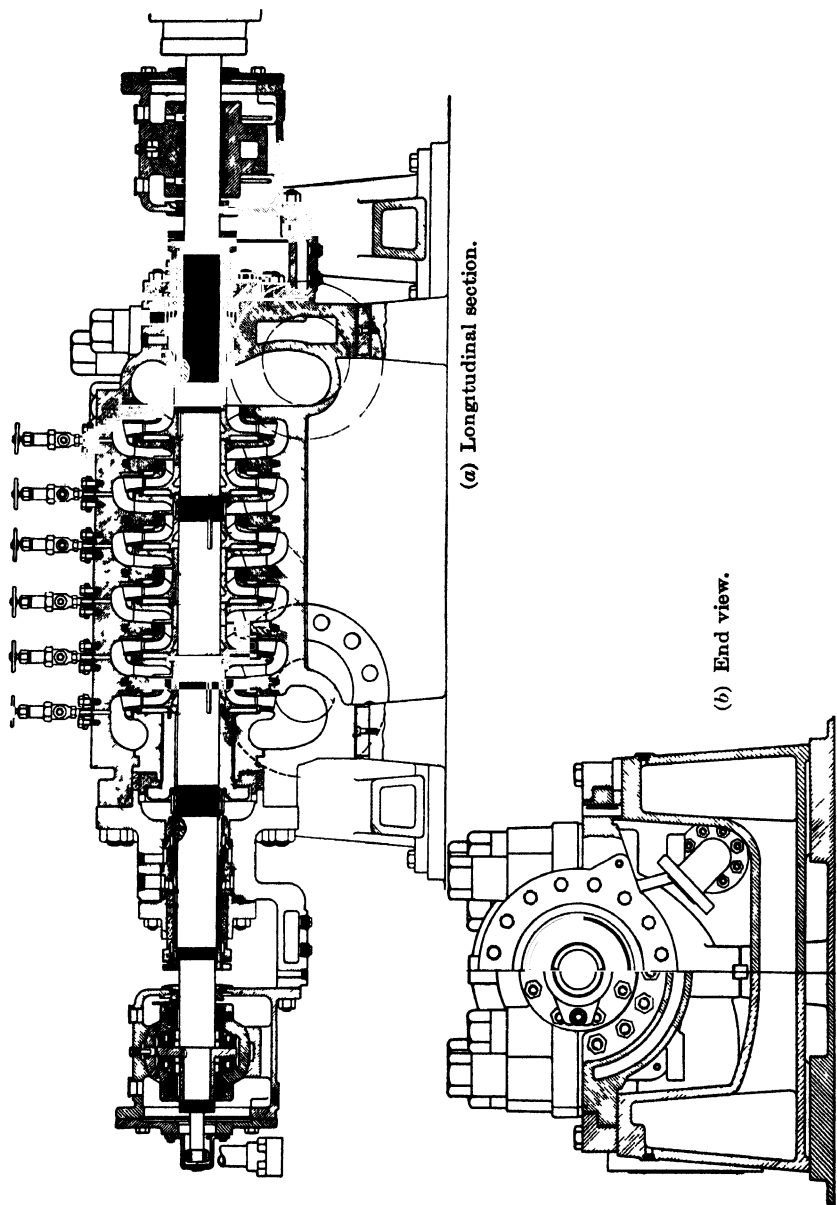



Fig 291 —Cameron six-stage, 1,500-lb. pressure centrifugal pump.

to neutralize end thrust. The stuffing boxes are surrounded by water under pressure. This feature prevents inward leakage of air, which is important when handling boiler feedwater.

Figure 291 illustrates the construction of a six-stage, turbine pump used for pumping hot water at pressures up to 1,500 lb. per square inch.

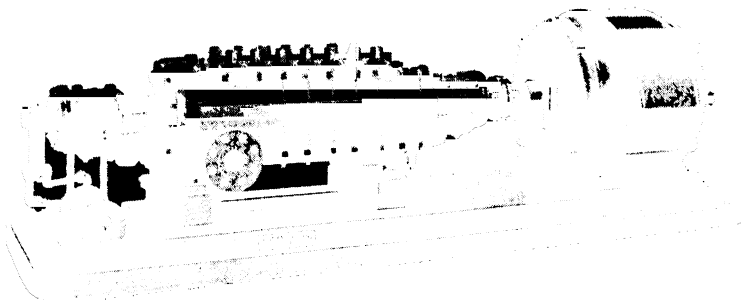



FIG 292.—Multi-stage, high-pressure centrifugal pump (Cameron).

Closed impellers with diffusion rings are used in all stages except the final stage, where a volute chamber is provided. Beyond the high-pressure stage, there is a balancing drum keyed to the shaft. Thrust in either direction is neutralized through the action of the water on this drum. A small auxiliary pump supplies oil under pressure to a Kingsbury thrust bearing. Figure 292 shows a photograph of a multi-stage pump built for the same service.

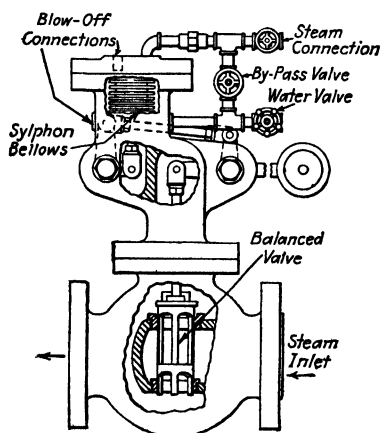



FIG. 293.—Copes pump governor.

Characteristic curves of a centrifugal pump, as illustrated in Fig. 269, show the relation of head, power and efficiency to the capacity. The curves shown are taken from test results of a pump operating at constant speed.

A pump of this type should be operated at the conditions at or near maximum efficiency, for best results. To facilitate governing, a centrifugal pump operating at constant speed should show a gradually increasing head, with maximum head at zero capacity.

281. Pump Governors.—Steam-driven pumps may be automatically regulated to give a constant discharge pressure regardless

of the rate of flow. This is done by controlling the speed or rate of operation, which can be accomplished by throttling the steam entering the pump.

Figure 293 illustrates a type of pump governor in which steam from the main supply header enters the inside of a sylphon bellows. The outside is subjected to the pressure in the water-discharge line of the pump. Changes in pressure expand or contract the bellows, and this motion, through a system of bell cranks and connecting links, actuates the weighted lever, thus opening or closing the throttle valve in the steam line to the pump. The governor can be cut out of service by closing the steam valve and opening the water by-pass valve, which subjects the inside and outside of the bellows to the same pressure. In this case, the weighted arm opens wide the throttle valve. This governor may be used for centrifugal or reciprocating pumps.

282. Injectors.—Injectors are a type of jet pump used almost universally on locomotive and small portable boilers to pump or deliver water into a boiler under operation. While not efficient as a pump, it is compact, inexpensive and returns to the boiler, as condensate, all of the steam which is required for its operation. Injectors will not handle hot water, and they are, under certain conditions, not dependable. The maintenance cost is negligible since the only moving parts are check valves.

In the simplest form of injector, steam, at high velocity, enters a converging nozzle, termed the combining tube. Owing to the velocity of the steam an ejector effect is produced, and a vacuum is created at the entrance of the combining tube. Water from the supply is thence drawn into the tube; it is entrained by the steam and acquires a high velocity. This action is accompanied by a condensing of the steam which increases the vacuum and in turn increases the flow of water. The water, after acquiring velocity, leaves the combining tube and enters a diverging delivery tube. In this tube the kinetic energy of the water jet is converted into pressure, and the water is forced through a check valve into the boiler.

Figure 294 shows a sectional view of an automatic injector. In operation, the steam flows through the combining tube of this injector, opens the ring valve, and passes to the overflow line. The vacuum or suction created at the entrance of the combining tube draws water to the chamber surrounding the first or steam nozzle. This condenses the steam and causes additional vacuum, which closes the ring valve. The ring valve is essentially a check valve. The velocity of the water builds up, and this is converted into pressure in the delivery tube. Continuing the flow, the water opens the check valve in the discharge

line, and then goes freely to the boiler. The action, as described, is more or less simultaneous.

283. Hydraulic Units.—The U. S. gallon (gal.) and the cubic foot (cu. ft.) are the units used in this country for measuring liquids in

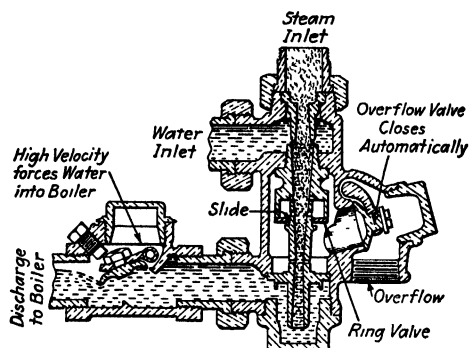



FIG. 294.—Penberthy automatic injector.

large bulk. The gallon contains 231 cu. in., which is equivalent to 0.134 cu. ft. The density of water is as shown in Table 14-1.

TABLE 14-1.—DENSITY OF WATER AT VARIOUS TEMPERATURES

Temperature, °F.	Density, lb. per cubic feet
32	62.42
50	62.42
70	62.3
90	62.11
110	61.88
130	61.54
150	61.20
170	60.79
190	60.39
210	59.92

Pressure of water may be measured as inches of mercury (in. Hg), pounds per square inch, ounces per square inch, feet of water, or inches of water. A pressure in one unit can be changed to an equivalent pressure in any other. Thus,

a. In. Hg. × 0.491 = lb. per square inch.

b. $\frac{h}{12} \times \frac{62.3}{144}$ or $0.036h$ = lb. per square inch.

in which

h = pressure, in inches of water at 70°F.

$$c. \quad H \times \frac{62.3}{144} \text{ or } 0.4326H = \text{lb. per square inch.}$$

in which

H = pressure, feet of water at 70°F.

Atmospheric pressure of 14.7 lb. per square inch is equal to a head of approximately 34 ft. of water at ordinary temperatures.

284. Nomenclature.—In reference to pumping, the *static head* is the total vertical distance through which the liquid is transported. With a liquid at rest, the height of the water level above the point at which the head is measured is the static head. The pressure head of a liquid flowing in a pipe is determined by making a connection to the pipe, at right angles to the direction of motion, and measuring with a suitable gage. This corresponds to the static head in air measurements. The pressure head indicates the pressure tending to burst the pipe. Pressure head on a pump is made up of the *discharge head* and *suction head.* Discharge head is the pressure of discharge, measured at the pump, on a level with the center line. The *suction head* is less than or greater than atmospheric pressure, depending on whether the supply tank is below or above the center line of the pump. The *velocity head* depends upon the velocity of flow and may be determined for both the suction and discharge of a pump. The *net velocity head* is the difference between that in the discharge and that in the suction line. The *total dynamic head* is equal to the sum of the pressure heads and velocity heads, while the pump is operating. The total dynamic head can be measured (if the pressures are small) by inserting a Pitot tube in the pipe, with the open end pointed against the direction of flow. The height of the water column equivalent to the pressure in the Pitot tube is the total dynamic head.

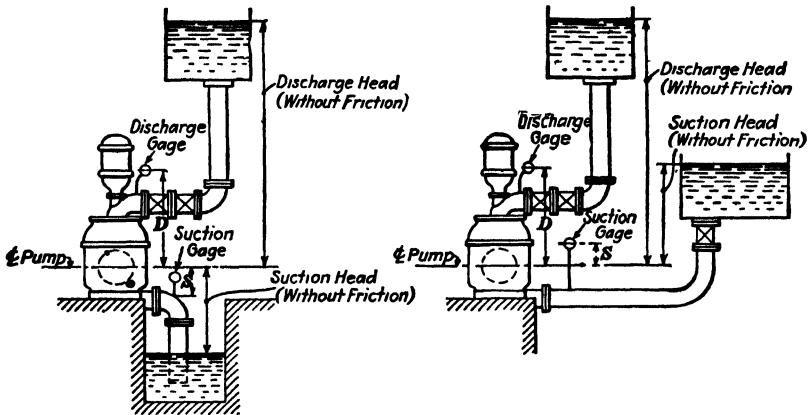
If gages are used for measuring the pressure head in the discharge and suction pipe, they should be placed as close as possible to the pump. A discharge pressure gage indicates the pressure at the level of the center of the gage, and the suction pressure gage indicates the pressure at the point where it connects to the suction pipe; that is, when the pressure in the suction pipe is less than that of the atmosphere. If the pressure is greater than atmospheric, the suction pressure gage indicates the pressure at the level of the center of the gage. Figure 295 illustrates these points.

285. Dynamic Head.—The following equations show the method used in calculating the total dynamic head. For a pump with suction lift, see Fig. 295 *a*.

$$h_t = h_D \pm D + h_s + S + \left(\frac{V_D^2}{2g} - \frac{V_s^2}{2g}\right) \qquad (155)$$

in which

h_t = total dynamic pressure head, ft.
h_D = discharge pressure head, ft.
h_s = suction pressure head, ft.
V_D = velocity in discharge pipe, ft. per second.
V_s = velocity in suction pipe, ft. per second.
D = vertical distance of discharge gage above (+) or below (−) pump center line, ft.
S = vertical distance of point where suction gage is connected with suction pipe, below center line of pump.

For a pump with positive suction head, see Fig. 295 *b*.

a. With supply tank below pump intake. *b.* With supply tank above pump intake.
FIG. 295.—Illustrating method of measuring heads on a pump.

$$h_t = h_D \pm D - h_s \pm S + \left(\frac{V_D^2}{2g} - \frac{V_s^2}{2g}\right) \qquad (156)$$

in which all symbols are as above, except S is measured the same as D.

It should be noted that if the discharge pressure gage is above the pump center line, D is positive, and if below, it is negative in both equations. When the suction gage is above the pump center line, in Eq. (156), S is negative, and if below, it is positive.

The head, either discharge or suction, may be measured by gages in units of pounds per square inch. In this case the head must be changed to its equivalent in feet of water. The head may also be determined by measuring directly the vertical distance, in feet, from the center line of the pump to the water level in the supply tank or

in the discharge stand pipe, provided, of course, that the water surface is open to the atmosphere. The most accurate measurement, however, is obtained at the pump, by means of gages, as this takes into account the friction of flow.

For reciprocating pumps, the velocity heads are often omitted, as the velocities are low and the difference between the velocity heads in discharge and suction pipes is small. The velocity head may be calculated from the following equations:

$$h_{vD} = \frac{V_D^2}{2g}$$

$$V_D = \frac{Q}{A_D} \tag{157}$$

in which

h_{vD} = velocity head in discharge pipe, ft. of water.
V_D = velocity in discharge pipe, ft. per second.
Q = water discharged, cu. ft. per second.
A_D = inside area of discharge pipe, sq. ft. (see Table 14-3).

Similarly, the velocity head in the suction pipe is calculated.

In order to prevent excessive friction loss, discharge velocities are commonly kept below 8 ft. per second for reciprocating pumps, and below 12 ft. per second for centrifugal pumps.

286. Water Horsepower.—The water horsepower of a pump depends on the weight of water actually delivered and the total dynamic head. Thus

$$\text{water hp.} = \frac{Wh_t}{33{,}000} \tag{158}$$

in which

W = water delivered, lb. per minute.
h_t = total head, ft. of water.

287. Indicated Horsepower.—The indicated horsepower of either the steam cylinder or the water cylinder of a reciprocating pump is determined by the expression used for a steam engine cylinder, which is

$$\text{i.hp.} = \frac{PLAN}{33{,}000} \tag{159}$$

in which

P = indicated mean effective pressure, lb. per square inch.
L = length of stroke, ft.
A = net area of piston, sq. in.
N = number of effective strokes per minute.

This gives the indicated horsepower for one end of the cylinder. The mean effective pressure is determined from an indicator diagram.

288. Mechanical Efficiency and Brake Horsepower.—The mechanical efficiency of a reciprocating pump is the ratio of the indicated power in the water end to that in the steam end.

$$\text{mechanical efficiency} = \frac{\text{i.hp. (water)}}{\text{i.hp. (steam)}} \times 100 \qquad \text{per cent} \quad (160)$$

The brake horsepower is used in connection with power pumps driven by electric motors or steam turbines and is not applied to direct-acting reciprocating pumps. The brake horsepower is the power actually delivered by the prime mover. In this case,

$$\text{mechanical efficiency} = \frac{\text{i.hp. (water)}}{\text{b.hp.}} \times 100 \qquad \text{per cent} \quad (161)$$

289. Volumetric Efficiency of a Pump.—Volumetric efficiency is the ratio of the quantity of water actually delivered to the piston displacement, both expressed in the same units, usually in cubic feet per minute. The *slip* of a pump is the difference between the piston displacement and the quantity of water actually delivered. Slip is expressed either as cubic feet per minute, or as percentage of piston displacement.

290. Pump Thermal Efficiency and Duty.—Thermal efficiency may be expressed in two ways: as the ratio of the heat equivalent of the water horsepower to the heat supplied, or as the duty, which is the number of foot-pounds of work done by the pump per million B.t.u. consumed by the driving engine. Thus

$$\text{thermal efficiency} = \frac{2{,}545 \times \text{water hp.}}{W_s(h_1 - h_{f2})} \times 100 \qquad \text{per cent} \quad (162)$$

$$\text{duty} = \frac{\text{water hp.} \times 33{,}000 \times 60}{W_s(h_1 - h_{f2})} \times 1{,}000{,}000 \qquad (163)$$

in which

W_s = steam consumed by the pump, lb. per hour.

h_1 = enthalpy of initial steam, B.t.u. per pound.

h_{f2} = enthalpy of liquid at exhaust pressure, B.t.u. per pound.

The duty may also be calculated as the foot-pounds of work done per thousand pounds of dry steam used.

291. Water Measurement.—The most accurate method of measuring water delivered by a pump is by volume, or by actually weighing the water delivered in a definite time interval. The latter is possible only with comparatively small quantities of water. In tests of pump-

ing equipment, the water delivered is usually measured by the use of a weir, orifice, Pitot tube or a Venturi gage. The error of measurement in using any one of these methods is small enough to be negligible.

In using a *weir*, the water is pumped into a tank with suitable baffles provided to eliminate currents caused by the discharge of the pump. There should be no noticeable velocity of water toward the weir or the determinations will be in error. A weir is a notch, usually rectangular or triangular (V notch, see Fig. 148), in the vertical side of the tank, through which the water is made to flow. In a rectangular weir the *crest* is the lower edge over which the water flows. A thin metal plate with the outer edge beveled to an angle of from 30 to 60 deg. is used for the crest and the sides.

The height of water over the crest is measured by a hook gage located back from the weir to avoid the effect of the curvature of the surface as the liquid approaches the weir. The zero of the hook gage is set as the same level as the crest.

If both sides of the weir are some distance from the sides of the tank, both sides of the stream are fully contracted, and the weir is said to have *end contractions.* A weir with both edges coinciding with the sides of the tank is said to be without end contractions, and the water overflows without being deflected by the edges of the weir.

The most widely used formulas for rectangular weirs are the following, derived by James B. Francis:

$$Q = C5.347(b - 0.1H)H^{3/2} \text{ (for one end contracted)} \quad (164)$$

$$Q = C5.347(b - 0.2H)H^{3/2} \text{ (for both ends contracted)} \quad (165)$$

$$Q = C5.347bH^{3/2} \text{ (without end contractions)} \quad (166)$$

in which

Q = cubic feet of water per second.

b = length of weir, ft.

H = head over crest, ft.

C = coefficient of discharge, from Table 14-2.

In Eqs. (164), (165) and (166) no account is taken of the head due to the velocity of the water approaching the weir. For accurate work this velocity head is included in the Francis formulas. As these formulas were derived from experiments on long weirs, a coefficient of discharge is used with small weirs. The coefficient of discharge is given in Table 14-2.

For the V-notch weir with a 90-deg. angle, the formula for discharge as derived by Prof. James Thompson is given by Eq. (167). Equation (168) is used for any angle between sides.

$$Q = 2.544H^{5/2} \quad (167)$$

in which

$$Q = 2.504\left(\tan \frac{B}{2}\right)^{0.996} \times H^{2.47} \qquad (168)$$

Q = cubic feet per second.
H = head above the apex of the triangle, ft.
B = angle between sides, deg.

A *Pitot tube,* for measuring pressures, is placed in a pipe with the opening turned in the direction against the flow of water. Placed in

TABLE 14-2.—COEFFICIENT OF DISCHARGE OF WEIRS, FRANCIS FORMULA*

Heads on weir, ft.	Length of weir, ft.			
	0.5	1	2	3
0.1	0.629	0.639	0.646	0.652
0.15	0.616	0.625	0.634	0.638
0.2	0.608	0.618	0.626	0.630
0.25	0.602	0.612	0.621	0.624
0.3	0.598	0.608	0.616	0.619
0.35	0.594	0.604	0.612	0.616
0.4	0.592	0.601	0.609	0.613

* MERRIMAN, "Treatise on Hydraulics."

the center of the pipe, it measures the velocity head at the center only. The ratio between the center velocity and average velocity may be determined, as follows:

$$V_a = 0.84\sqrt{2gh_v} \qquad (169)$$

in which

$0.84 = \frac{V_a}{V_c}$ (approximate).
V_a = average velocity, ft. per second.
h_v = velocity head at center of pipe, ft. of water.
V_c = velocity at center, ft. per second.

Then, the quantity of flow

$$Q = CAV_a \qquad (170)$$

in which

Q = discharge, cu. ft. per second.
C = Pitot nozzle coefficient (0.95 to 0.98).
A = area of pipe, sq. ft.

Example 14-1.—The following data were taken on a test of a duplex, direct-acting water pump:

Pump Data: pipe diameters, discharge, 6 in., suction, 8 in.; cylinder diameters, steam, 14 in., water, 8 in ; piston-rod diameter, 1.93 in.; average length of stroke,

TABLE 14-3.—WROUGHT-IRON WELDED STEAM, AIR, GAS, AND WATER PIPE (Standard Dimensions of National Tube Works Company)

Diameter				Circumference		Transverse areas	
Nominal internal, in.	Actual external, in.	Approx. internal, in.	Thickness, in.	External, in.	Internal, in.	External, sq. in.	Internal, sq. in.
1/8	0.405	0.27	0.068	1.272	0.848	0.129	0.0573
1/4	0.540	0.364	0.088	1.696	1.144	0.229	0.1041
3/8	0.675	0.494	0.091	2.121	1.552	0.358	0.1917
1/2	0.840	0.626	0.109	2.630	1.957	0.554	0.3048
3/4	1.050	0.824	0.113	3.209	2.589	0.866	0.5333
1	1.315	1.050	0.134	4.131	3.292	1.358	0.8626
1 1/4	1.660	1.380	0 14	5.215	4.335	2.164	1.496
1 1/2	1.900	1.611	0.145	5.960	5.061	2.835	2.038
2	2.375	2.067	0 154	7.461	6.494	4.43	3.356
2 1/2	2.875	2.468	0 204	9.032	7.758	6.492	4.784
3	3.500	3.067	0 217	10.996	9.636	9.621	7.388
3 1/2	4.000	3.548	0 226	12.566	11.146	12.566	9.887
4	4.500	4.026	0.237	14.137	12.648	15.904	12.73
4 1/2	5.000	4.508	0 246	15.708	14.162	19.635	15.691
5	5.563	5.045	0.259	17.477	15.849	24.306	19.99
6	6.625	6.065	0 28	20.813	19.054	34.472	28.888
8	8.625	7.982	0.322	27.096	25.076	58.426	50.04
10	10.750	10.019	0.366	33.772	31.477	90.763	78.839

9.5 in.; double strokes per minute, 56; steam consumed, 834 lb. per hour; average steam m.e.p., H.E., 14.71 lb. per square inch.; C.E., 15.00 lb. per square inch; average water m.e.p., H.E., 42.5 lb. per square inch, C.E., 45.8 lb. per square inch.

Pressures and temperatures: pressure in steam line, 142 lb. per square inch gage; quality in steam line, 98 per cent; pressure below throttle, 18 lb. per square inch gage; exhaust, 7.06 in. of mercury, vacuum; discharge water, 40 lb. per square inch, gage; vacuum in suction pipe, 13.6 in. of mercury; barometer, 29.46 in. of mercury; temperature, room, 72°F.; discharge water, 77°F.

Miscellaneous Data: height of pump center line above suction-gage connection, 1.4 ft.; height of pump center line above discharge gage, 1.27 ft.; width of weir, rectangular, 1.5 ft.; height of water over crest, 0.35 ft.; end contractions, 2. Zero of hook gage at same level as crest of weir.

Required: (*a*) water discharged, cubic feet per second; (*b*) water horsepower; (*c*) mechanical efficiency; (*d*) thermal efficiency; (*e*) duty, foot-pounds per million B.t.u.

Solution.—*a*. Water discharged, cubic feet per second:
From Eq. (165),

$$Q = C \times 5.347(1.5 - 0.2 \times 0.35)0.35^{3/2}$$
$$Q = 0.608 \times 5.347 \times 1.43 \times 0.207 = 0.96 \text{ cu. ft. per second}$$

Coefficient C from Table 14-2.

b. **Water horsepower:**

$$\text{Area of discharge pipe (Table 14-3)} = \frac{28.888}{144} = 0.2 \text{ sq. ft.}$$

$$\text{Area of suction pipe} = \frac{50.04}{144} = 0.347 \text{ sq. ft.}$$

$$V_D = \frac{0.96}{0.2} = 4.8 \text{ ft. per second (discharge)}$$

$$V_s = \frac{0.96}{0.347} = 2.76 \text{ ft. per second (suction)}$$

Head due to velocity in pipe

$$h = \frac{4.8^2}{2 \times 32.2} = 0.358 \text{ ft. (discharge)}$$

$$h = \frac{2.76^2}{2 \times 32.2} = 0.118 \text{ ft. (suction)}$$

From Eq. (155),

$$h_t = 40 \times 2.315 - 1.27 + 13.6 \times 0.491 \times 2.315 + 1.4 + 0.358 - 0.118 = 106.87 \text{ ft.}$$

Using Eq. (158),

$$\text{water hp.} = \frac{0.96 \times 62.26 \times 60 \times 106.87}{33{,}000} = 11.58$$

c. Mechanical efficiency:

Average indicated horsepower, steam cylinder

$$\text{H.E.} = \frac{14.71 \times 9.5 \times 154 \times 56}{12 \times 13{,}000} = 3\ 04$$

$$\text{C.E.} = \frac{15 \times 9.5 \times 151 \times 56}{12 \times 33{,}000} = 3\ 04$$

Total, one cylinder = 6.08

Total indicated horsepower, steam cylinders = 12.16

Average indicated horsepower, water cylinder

$$\text{H.E.} = \frac{42.5 \times 9.5 \times 50.2 \times 56}{12 \times 33{,}000} = 2.86$$

$$\text{C.E.} = \frac{45.8 \times 9.5 \times 47.2 \times 56}{12 \times 33{,}000} = 2.91$$

Total, one cylinder = 5.77

Total indicated horsepower, water cylinders = 11.54

$$\text{Mechanical efficiency} = \frac{11.54}{12.16} \times 100 = 95 \text{ per cent}$$

d. Thermal efficiency:

$$\text{B.t.u. per hour supplied} = 834(333.84 + 0.98 \times 861.6 - 167.8) = 844{,}000$$

$$\text{Thermal efficiency} = \frac{11.58 \times 2{,}545}{844{,}000} \times 100 = 3.5 \text{ per cent}$$

e. Duty:

$$\text{Duty} = \frac{11.58 \times 33{,}000 \times 60}{844{,}000} \times 1{,}000{,}000$$

Problems

1. Change the following pressures to equivalent values expressed as pounds per square inch: 2.5 inches of mercury; 2.5 ft. of water; 2.5 in. of water; 20 oz. per square inch; 2.5 lb. per square foot; 35 ft. of water.

2. Change the following pressures to equivalent values expressed as feet of water: 6.75 lb. per square foot; 155 lb. per square inch; 22.5 inches of mercury.

3. A reciprocating pump delivers 150 gal. of water per minute, at 60°F., against a discharge pressure of 50 lb. per square inch. The center of the discharge gage is 1.5 ft. below the pump center line. Pipe sizes are: discharge 6 in., suction 8 in. The vacuum in the suction pipe is 4.5 inches of mercury, and the gage is attached 1 ft. below the pump center line. Calculate the total head.

4. Change the data of Problem 3, so that there is a positive suction pressure of 5 lb. per square inch, when the center of the suction gage is 1 ft. above the pump center line, and calculate the total head.

5. A boiler feed pump delivers 200 gal. of water at 160°F., per minute. The pump uses 65 lb. of steam per indicated horsepower hour, at 150 lb. per square inch absolute pressure; quality 0.985; exhaust pressure 17.5 lb. per square inch absolute. The steam indicated horsepower is 35, and the water pressure in the discharge pipe is 200 lb. per square inch gage; the pressure gage is 1.75 ft. above the pump center line; vacuum in the suction line is 6 in. of mercury; the suction gage is attached 2 ft. below the pump center line. Pipe sizes are: suction 8 in.; discharge 6 in. Calculate (*a*) the water horsepower, (*b*) the thermal efficiency, and (*c*) the duty per million B.t.u.

6. A boiler feed pump delivers 350 gal. of water per minute at 185°F., using 87 lb. of dry steam per indicated horsepower hour, at 200 lb. per square inch absolute pressure, and exhausting at 20 lb. per square inch absolute. The steam indicated horsepower is 55. Pipe sizes are as follows: suction 10 in., discharge 8 in. The water pressure in the discharge pipe is 215 lb. per square inch gage; the discharge gage is 2 ft. below the pump center line. The positive suction pressure is 10 lb. per square inch, and the suction gage is 2 ft. below the pump center line. Calculate (*a*) the water horsepower, (*b*) the thermal efficiency, and (*c*) the duty based on 1,000,000 B.t.u.

7. A simplex, direct-acting pump has the following water-cylinder dimensions: diameter 9 in.; stroke 15 in. Pump operates at 39.7 double strokes per minute. The water discharged runs over a rectangular weir with two contractions; length of weir 6 in.; the head over the crest is 6.4 in. Calculate the slip of the pump in cubic feet per minute, and also as a percentage of the piston displacement.

8. A duplex, direct-acting pump with water-cylinder diameter of 12-in. and 20-in. stroke operates at 54 double strokes per minute. The water delivered flows over a V-notch weir, the head on the weir being 14.5 in. Calculate the slip in cubic feet per minute, and in percentage of the piston displacement.

9. On test, a Pitot tube is placed in the center of a pump discharge line 6 in. in diameter, and gives a reading of 13.5 ft. of water. Using a coefficient for the Pitot tube of 0.965, calculate the flow of water in gallons per minute.

10. A 10- by 10- by 12-in. simplex pump with 1¾-in. piston rod runs at 32 double strokes per minute. If the slip is 8 per cent and the water temperature is 80°F., calculate the water discharge, gallons per minute.

11. A pump test gives the following data: Water discharge 200 gal. per minute, temperature of water 130°F., discharge pipe 4 in., suction pipe 6 in. Calculate the difference between the velocity heads in discharge and suction pipes.

CHAPTER XV

INTERNAL COMBUSTION ENGINES

292. Historical Notes.—Previous to the year 1830 no practical results had been attained in the field of internal-combustion engines. There are, however, records from as early as the seventeenth century which describe crude experiments dealing with engines of this type. One interesting instance of this occurred in France, in 1678, when recognition was given for an engine which consumed gunpowder. This, like Newcomen's steam engine, involved a cylinder and piston and utilized the pressure of the atmosphere. During operation, hot gases from the burning powder filled the cylinder to maximum volume. Then, as a result of a suitable cooling effect, a partial vacuum was created, and the atmosphere, acting on the open side of the piston, furnished the pressure for the working stroke. About 1830 Samuel Brown, of England, developed a double-acting engine which operated on the same principle.

In 1860 J. J. E. Lenoir, of Paris, was accredited with the design of the first internal-combustion engine having commercial value. It was double-acting, with a crank and flywheel, used slide valves for controlling both admission and exhaust and operated on gas. This engine met with reasonable success and was used in various European countries.

Beau de Rochas of France, in 1862, proposed the four-stroke cycle now widely used in the design of internal-combustion engines. No practical application of this cycle was made until 14 yr. later, when Dr. Nicholas A. Otto, of Germany, utilized it in building his so-called silent engine. Because of this first practical application this cycle is often called the Otto cycle.

Previous to this Dr. Otto and A. Langen had built what they called a free-piston engine. The cylinder was vertical and very long, and the piston was attached to a heavy, rack-gear piston rod which meshed with a pinion on the power shaft. Gas burning in the cylinder furnished the pressure for raising the piston to the top of its stroke. The gases were then allowed to exhaust, and the piston, by force of gravity, moved downward, which constituted the power stroke. Through the rack, pinion and a suitable clutching device the power was converted to the rotary shaft and flywheel.

Other events of note in the development of internal-combustion engines include the invention, in 1874, of the liquid-fuel engine, by G. B. Brayton, an American, and the invention, in 1881, of the two-stroke cycle gas engine by Dugald Clerk of England. Brayton's engine was of particular importance as it consumed gasoline which at that time was a by-product from the manufacture of kerosene.

In all of these early engines the fuel ignition had been accomplished by means of an open flame. A more or less successful attempt to overcome the many difficulties encountered in the use of the flame ignition was made by Stuart Ackroyd, of England, when he introduced the "hot-bulb" or "surface-ignition" engine in 1892. This engine was designed for the four-stroke cycle, low compression, and accomplished ignition by spraying the fuel onto a hot metal surface within the cylinder. At about this same time Dr. Rudolf Diesel, of Germany, introduced what he called the "rational heat motor," which was to operate on coal dust. This engine was to have been started by an explosive introduced into the cylinder. The first attempt at starting proved destructive to the engine and failure of the design. Further experiments on the part of Dr. Diesel evolved an engine which operated on the present-day Diesel cycle. A year later, in 1898, the first Diesel engine to be manufactured in America made its appearance.

293. Modern Development.—Although the internal-combustion engine had its birth in the nineteenth century, its important development and application to almost every conceivable need have occurred since 1900. The most notable application, is of course, to vehicles. It is also interesting to note that approximately 16 per cent of all power consumed by industries in the United States is derived from engines of this type, and that nearly 80 per cent of all ships built during 1930, between the sizes of 8,000 and 25,000 tons, are equipped with Diesel engines.

294. Details of Internal-combustion Engines.—The various types of internal-combustion engines in service today derive their power direct from fuels in either the gaseous or liquid form. Engineers are, however, still experimenting with the use of powdered fuels, and there have been some very interesting results obtained toward this end.

Fuels for this purpose possess energy by virtue of their ability to evolve heat when ignited and burned in the presence of oxygen. If a fuel is of such quality that its burning is rapid, the heat produced effects a rapid rise in temperature of the resulting gases. This, in turn, causes a corresponding increase in pressure which is followed by expansion. If the time element involved is instantaneous the effect is an explosion, and this is what takes place to a greater or lesser extent in

the cylinder of an internal-combustion engine. In most cases the air and fuel are compressed previous to ignition. All engines of this type attempt to utilize the expansive force of the gases and convert it into energy which is readily usable.

The essential mechanism of an internal-combustion engine consists of a frame, cylinder, piston, valves, connecting rod, crank shaft, flywheel and bearings. Means for supplying fuel and air and for accomplishing ignition are also necessary. The materials used in the construction of the various parts include almost every conceivable metal and alloy used by industry, and there are attempts on every side to obtain the best and most economical material for each particular purpose.

The cylinders are usually cooled by means of water jackets, and the pistons are provided with packing rings. In the four-cycle engine, valves are necessary for controlling the inlet of fuel and air and the exhaust of the waste gases. In the two-cycle engine the exhaust and air-inlet port may be arranged so as to be opened and closed by the piston. Double-acting engines require a piston rod, crosshead and connecting rod, as in the steam engine. The most common design, however, is single acting, in which the trunk piston, omitting the piston rod and separate crosshead, is used.

The various features of internal-combustion engines will be taken up in the following articles.

295. Theoretical Cycles.—The theoretical cycles of operation for internal-combustion engines are based on assumptions that a frictionless piston is used in a cylinder having non-heat-conducting walls, and that the medium in the cylinder is air, to which heat may be added or rejected through the head of the cylinder. Fuel to provide heat is needed in the real engine cycle, but with the ideal engine, it is assumed that heat is supplied from a source of such magnitude that its temperature may remain constant regardless of the amount of heat abstracted. It is also assumed that there is a refrigerator of infinite capacity; that is, one which will absorb heat from the air, without effecting a change of temperature. A theoretical efficiency based upon the above assumptions is known as the *air standard efficiency.*

296. The Carnot Cycle.—In 1824 Sadi Carnot, a French engineer, described a reversible thermodynamic cycle to give the maximum possible efficiency between two temperatures, for any working medium. He assumed an ideal reversible engine; that is, one having the same efficiency when acting either as an engine or as a compressor. This cycle, on the pressure-volume diagram, is illustrated in Fig. 296, and consists of four changes of state, as shown. A source of heat S, at a

high temperature T_1, contains an infinite supply of heat; and a cold body R, of infinite heat-absorbing capacity, and an insulator I are used in the operation of the ideal engine. At the beginning of the cycle, the piston is at A_1, and the air in the cylinder is at a temperature T_1. The heat source S is placed against the cylinder head, causing the gas to expand at constant temperature (isothermal) as heat is absorbed. The piston moves to B_1. Then, S is replaced by the insulator I, and the expansion of the gas continues without addition or rejection of heat (adiabatic). The piston moves to C_1, and the temperature drops to T_2. When the end of the working stroke is reached the refrigerator R is placed at the cylinder head, and a rejection of heat at constant temperature (isothermal) takes place as the piston travels to D_1. This change is stopped at a point such that adiabatic compression, with the insulator I placed at the cylinder head, will bring the air back to its initial condition, as the piston returns to A_1. Thus, it may be seen that the cycle consists of two isothermal and two adiabatic changes of state.

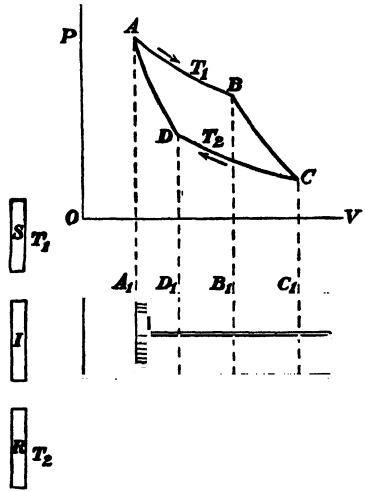



Fig. 296.—The Carnot cycle.

This series of operations, as shown, gives the work represented by the area $ABCD$. The cycle efficiency may be determined by first developing expressions for the heat absorbed per cycle Q_1 and the heat rejected Q_2. Thus

$$Q_1 = AwRT_1 \log_e \frac{V_b}{V_a}$$

$$Q_2 = AwRT_2 \log_e \frac{V_c}{V_d}$$

$$e = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

$$= 1 - \frac{AwRT_2 \log_e \frac{V_c}{V_d}}{AwRT_1 \log_e \frac{V_b}{V_a}}$$

There is no heat absorbed or rejected during the two adiabatic changes. The T-V relations are expressed by the following:

$$\left(\frac{V_c}{V_b}\right)^{k-1} = \frac{T_b}{T_c} = \frac{T_1}{T_2}$$

$$\left(\frac{V_d}{V_a}\right)^{k-1} = \frac{T_a}{T_d} = \frac{T_1}{T_2}$$

Hence

$$\frac{V_c}{V_b} = \frac{V_d}{V_a}$$

or

$$\frac{V_c}{V_d} = \frac{V_b}{V_a}$$

and

$$\log_e \frac{V_c}{V_d} = \log_e \frac{V_b}{V_a}$$

Therefore

$$e = 1 - \frac{T_2}{T_1} = \frac{T_1 - T_2}{T_1} \tag{171}$$

In the foregoing equations

e = Carnot cycle efficiency.
V = cylinder volumes, cu. ft.
T = absolute temperatures, °F.
R and k = constants for the gas.
Q = heat flow.
w = weight of gas, lb.

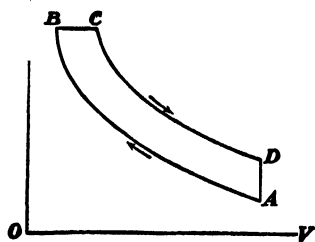



FIG. 297.—The ideal Diesel cycle.

297. The Diesel Cycle.—The Diesel cycle, drawn on the P-V plane, is illustrated in Fig. 297. It consists of a constant-pressure heat absorption BC, an adiabatic expansion CD, a constant-volume heat rejection DA and an adiabatic compression AB. The expressions, for heat flow in this cycle, are as follows:

$$Q_1 = wc_p(T_c - T_b)$$

$$Q_2 = wc_v(T_d - T_a)$$

$$e = 1 - \frac{Q_2}{Q_1} = 1 - \frac{wc_v(T_d - T_a)}{wc_p(T_c - T_b)}$$

$$= 1 - \frac{1}{k}\frac{T_d - T_a}{T_c - T_b} \tag{172}$$

For the constant-pressure and constant-volume changes the following relations are true:

$$T_c = T_b\left(\frac{V_c}{V_b}\right) \qquad \text{(constant pressure)}$$

$$T_d = T_a\left(\frac{P_d}{P_a}\right) \qquad \text{(constant volume)}$$

For the adiabatic changes:

$$\frac{P_a}{P_b} = \left(\frac{V_b}{V_a}\right)^k \tag{173}$$

$$\frac{P_d}{P_c} = \left(\frac{V_c}{V_d}\right)^k = \left(\frac{V_c}{V_a}\right)^k = \frac{P_d}{P_b} \tag{174}$$

Dividing Eq. (174) by Eq. (173),

$$\frac{P_d}{P_a} = \left(\frac{V_c}{V_b}\right)^k$$

Substituting this value of P_d/P_a in the constant-volume relation:

$$T_d = T_a\left(\frac{V_c}{V_b}\right)^k$$

Substituting values of T_d and T_c in Eq. (172),

$$e = 1 - \frac{T_a(V_c/V_b)^k - T_a}{k[T_b(V_c/V_b) - T_b]}$$

$$e = 1 - \frac{T_a}{T_b}\,\frac{\left(\frac{V_c}{V_b}\right)^k - 1}{k\left[\left(\frac{V_c}{V_b} - 1\right)\right]} \tag{175}$$

in which

e = efficiency of Diesel cycle

and other symbols are as given in the foregoing article.

In Eq. (175), in the place of the ratio of the temperatures at the beginning and end of compression, the ratios of the pressures or volumes can be used. Thus

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{k-1} = \left(\frac{P_a}{P_b}\right)^{\frac{k-1}{k}}$$

The *air-standard thermal efficiency* of the Diesel cycle depends on the compression ratio, and also on the ratio of the cylinder volume at the end of combustion to the clearance volume.

FIG. 298.—The ideal Otto cycle.

298. The Otto Cycle.—The Otto cycle is illustrated in Fig. 298. This cycle consists of a constant-volume heat absorption BC, and an adiabatic expansion CD, a constant-volume heat rejection DA and an adiabatic compression AB. The heat absorbed, the heat rejected

during the cycle and the cycle efficiency may be expressed as follows:

$$Q_1 = wc_v(T_c - T_b)$$
$$Q_2 = wc_v(T_d - T_a)$$
$$e = 1 - \frac{Q_2}{Q_1} = 1 - \left(\frac{T_d - T_a}{T_c - T_b}\right) \tag{176}$$

The adiabatic changes of state are expressed in the following relations:

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{k-1}$$
$$\frac{T_d}{T_c} = \left(\frac{V_c}{V_d}\right)^{k-1}$$

Since $V_b = V_c$ and $V_a = V_d$,

$$\frac{T_a}{T_b} = \frac{T_d}{T_c}$$

or

$$\frac{T_c}{T_b} = \frac{T_d}{T_a}$$

Then

$$T_d = T_a\left(\frac{T_c}{T_b}\right) \tag{177}$$

and

$$T_c = T_b\left(\frac{T_d}{T_a}\right) \tag{178}$$

Substituting the values of T_d and T_c from Eqs. (177) and (178) in Eq. (176) gives the following:

$$e = 1 - \frac{T_a\frac{T_c}{T_b} - T_a}{T_b\left(\frac{T_d}{T_a}\right) - T_b} = \frac{T_a\left[\left(\frac{T_c}{T_b}\right) - 1\right]}{T_b\left[\left(\frac{T_d}{T_a}\right) - 1\right]} = 1 - \left(\frac{T_a}{T_b}\right) \tag{179}$$

also

$$e = 1 - \left(\frac{P_a}{P_b}\right)^{\frac{k-1}{k}} = 1 - \left(\frac{V_b}{V_a}\right)^{k-1} = 1 - \frac{1}{r^{(k-1)}} \tag{180}$$

The *air standard efficiency* of the Otto cycle depends upon the pressure or temperature before and after compression and the compression ratio ($r = V_a/V_b$). The following example illustrates the use of the foregoing equations.

Example 15-1.—The pressure in an engine cylinder is 14.7 lb. per square inch absolute at the beginning of compression and the temperature is 150°F.; the maximum pressure attained inside the cylinder is 450 lb. per square inch absolute; the temperature at the end of combustion is 2400°F. Determine the air standard efficiency and the clearance in percentage of the piston displacement, for (*a*) the Diesel cycle; (*b*) the Otto cycle.

Solution.—*a*. Referring to diagram (Fig. 297).
If

$$V_p = \text{piston displacement volume}$$
$$V_b = mV_p$$

where

$$m = \text{clearance, as a decimal part of } V_p$$
$$V_a = V_b + V_p = (m+1)V_p$$

From the adiabatic compression

$$\frac{450}{14.7} = \left(\frac{V_a}{V_b}\right)^k = \left[\frac{V_p(m+1)}{mV_p}\right]^{1.4}$$
$$m = 0.952 \text{ or } 9.52 \text{ per cent}$$

From the adiabatic compression

$$T_b = T_a\left(\frac{P_b}{P_a}\right)^{\frac{k-1}{k}} = 610\left(\frac{450}{14.7}\right)^{0.286} = 1{,}625 \text{ deg. abs.}$$

Using Eq. (175), with T_c/T_B substituted for V_c/V_B:

$$e = 1 - \frac{610}{1{,}625}\,\frac{\left(\frac{2{,}860}{1{,}625}\right)^{1.4} - 1}{1.4\left(\frac{2{,}860}{1{,}625} - 1\right)}$$
$$e = 0.57, \text{ or } 57 \text{ per cent.}$$

b. Referring to diagram (Fig. 298).
From Eq. (4),

$$\frac{P_cV_b}{T_c} - \frac{P_aV_a}{T_a}$$

Since $V_b = V_c$,

$$\frac{V_b}{V_a} = \frac{14.7 \times 2{,}860}{450 \times 610} = 0.1532$$
$$\frac{V_b}{V_a} = \frac{m}{1+m} = 0.1532$$
$$m = 0.1809, \text{ or } 18.09 \text{ per cent}$$
$$e = 1 - 0.1532^{0.4} = 0.528, \text{ or } 52.8 \text{ per cent}$$

299. The Dual (or Combined) Cycle.—The dual cycle illustrated in Fig. 299 may be considered as a combination of the Otto and Diesel cycles. Heat absorption occurs partly at constant volume from B to C, and partly at constant pressure from C to D. Compression A to B

and expansion D to E are reversible adiabatic processes. Heat rejection from E to A is at constant volume.

The air standard efficiency for this cycle may be developed as follows:

$$Q_1 = {}_bQ_c + {}_cQ_d = wC_v(T_c - T_b) + wC_p(T_d - T_c)$$
$$Q_2 = {}_eQ_a = wC_v(T_e - T_a)$$
$$e = 1 - \frac{Q_2}{Q_1} = 1 - \frac{C_v(T_e - T_a)}{C_v(T_c - T_b) + C_p(T_d - T_c)} \tag{181}$$

Expressing all temperatures in terms of T_a:

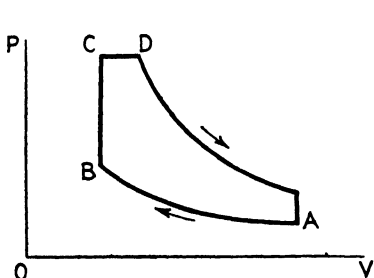



Fig. 299.—The ideal dual or combined cycle.

$$T_b = T_a\left(\frac{V_a}{V_b}\right)^{k-1}$$
$$T_c = T_b\left(\frac{P_c}{P_b}\right) = T_a\left(\frac{V_a}{V_b}\right)^{k-1}\left(\frac{P_c}{P_b}\right)$$
$$T_d = T_c\left(\frac{V_d}{V_c}\right) = T_a\left(\frac{V_d}{V_b}\right)\left(\frac{V_a}{V_b}\right)^{k-1}\left(\frac{P_c}{P_b}\right)$$
$$T_e = T_d\left(\frac{V_d}{V_e}\right)^{k-1} = T_a\left(\frac{V_d}{V_b}\right)^{k}\left(\frac{P_c}{P_b}\right)$$

Substituting for temperatures, the above values, using the symbol r (ratio of compression) for V_a/V_b, and simplifying, the efficiency becomes

$$e = 1 - \frac{(V_d/V_b)^k - (P_b/P_c)}{r^{k-1}\left[1 - \frac{P_b}{P_c} + k\left(\frac{V_d}{V_b} - 1\right)\right]} \tag{182}$$

The air standard efficiency of the dual cycle depends upon the following variables: V_a/V_b, the ratio of compression; V_d/V_b, the ratio of the volume after combustion to that at the start of combustion; and P_b/P_c, the ratio of the pressure at the start of combustion to that at the end of combustion. The efficiency increases as the ratio V_a/V_b increases, and similarly with an increase in the ratio P_b/P_c. However, an increase in the ratio V_d/V_b results in a decrease in the value of the air standard efficiency. This means that an increase in load will lower the efficiency.

300. Actual Realization.—The cycle efficiencies developed in the preceding articles are based on ideal engines following reversible cycles, as explained in detail in the discussion of the Carnot cycle. Actual engines designed to use these cycles as close as practicable have certain inescapable variations therefrom which reduce the actual

efficiencies as compared with the theoretical. For instance, a part of each cycle must be used in admitting a charge of fuel and combustion and in discharging from the cylinder the gaseous products of combustion after their expansion to low pressure. The compression ratio, or expansion ratio in the actual engine, is lower, for practical reasons, than in the ideal engine. Consequently, a large quantity of energy is lost as high-temperature gas is discharged from the engine. To use more of this energy would require a longer expansion, which would mean greater cylinder volume. The limitation on large cylinders is the increased cost of the engine. The difficulties in attaining perfect fuel combustion and perfect piston and cylinder lubrication present obstacles in reaching comparatively high efficiencies.

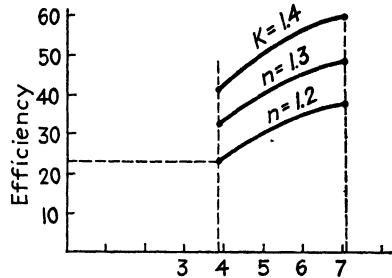



FIG. 300.—Effect of value of n on efficiency, Otto cycle.

The necessity of cooling the cylinder is also a factor in lowering the practical efficiency. Because of this, energy is lost by heating the cooling water, and the expansion and compression curves are not reversible adiabatics but follow the general curve $PV^n = C$. The effect of common values of n on otherwise theoretical efficiencies for the Otto cycle are shown in Fig. 300, with efficiency and compression ratio as ordinates.

The engines following the Otto cycle will give, on the average, actual thermal efficiencies of 45 per cent of theoretical results. Those following the Diesel cycle, because of the higher compression ratios possible, attain 60 per cent of theoretical efficiencies. Average economy results are as follows:

Type of engine	Fuel per b.hp.-hr., lb.	Expansion ratios
Automotive	0.55 to 0.60	5 to 5.5
Aviation	0.52	6.5
Farm engines	0.7	4

In comparing the Otto and Diesel cycles, it may be noted that the air standard efficiency of the Otto cycle increases with the compression ratio (the ratio of the volume before compression to that at the end of compression). A comparison of the efficiencies of the two cycles is shown in Fig. 301. Those for the Diesel cycle are based on a temperature increase of 2000°F. during combustion. With the same

compression ratio, the Otto cycle gives the higher efficiency. However, the Otto engine operates at a low-compression ratio to prevent preignition, while the Diesel engine compresses only air, is not limited and operates at a comparatively high-compression ratio.

301. Actual Operating Cycles.—The internal-combustion engine of either cycle must, in its actual cycle of operation, include the admission of fuel and air into the cylinder, combustion of the fuel, and the discharge of the burned gases at low pressure. These operations must occur in proper sequence, and must recur continually, if the engine is to do work over a period of time. Either the Diesel or the Otto cycles, as briefly outlined here, may be carried out in the two or four strokes.

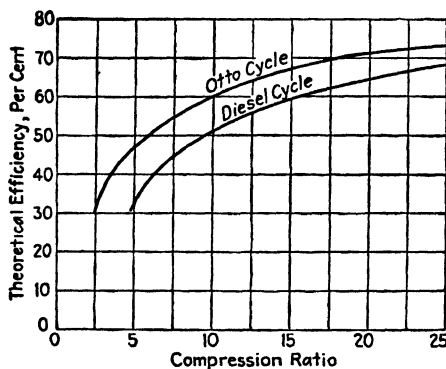



FIG. 301.—Curves showing comparison of the efficiency of the Otto and Diesel cycles.

The *two-stroke cycle* is completed in two strokes of the engine piston or one revolution of the crank shaft. In Fig. 302 *a* are shown the indicator and crank diagrams for an engine which operates on the Otto, two-stroke cycle. At the beginning of the power stroke, the piston is at the head-end dead center, and gases, at high pressure and temperature, fill the clearance space. The gases expand from A to B as the piston moves, until the exhaust port is opened and exhaust of the gases from the cylinder begins. On the diagram shown, the exhaust is represented by the short line $BB'C$. The power stroke is thus divided into two phases, expansion and exhaust. At the beginning of the return stroke a volume of fuel and air at a pressure above that of the atmosphere enters the cylinder and aids in expelling the remaining burned gases. After both ports close, the fuel and air mixture is compressed from D to E, and at near the dead-center position of the piston, ignition is effected and the fuel is burned. The admission and scavenging operations occur along the lines $B'CB'$ and the compression from D to E. From E to A is the constant-volume combus-

tion that completes the cycle. The compression stroke includes the phases of admission, scavenging, compression and combustion.

The four-stroke cycle (Fig. 302 *b*) requires four strokes of the piston or two revolutions of the crank shaft to complete the necessary operations. On the diagram shown, the expansion of the gases during the power stroke continues for practically the entire stroke. On the second stroke, the gases are discharged at a pressure slightly above that of the atmosphere, and during the suction stroke, a fresh charge of fuel and air is drawn into the cylinder, at a pressure slightly less than

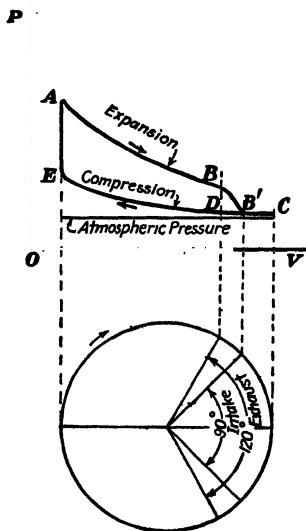



Fig. 302 *a*.—Indicator and crank diagrams for two-stroke cycle engine.

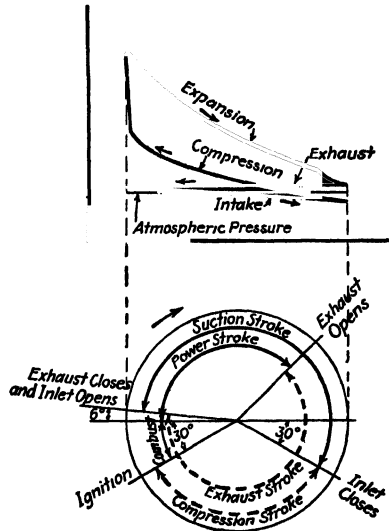



Fig. 302 *b*.—Indicator and crank diagrams for four-stroke cycle engine.

that of the atmosphere. The fourth stroke is the compression stroke with ignition and combustion occurring just before the dead-center position.

Each cylinder of a two-stroke cycle engine should develop twice the power of the four-stroke cycle cylinder of the same size and with the shaft rotating at the same speed. However, the former has imperfect scavenging, and, because there is a considerable volume of burned gas left in the cylinder at the start of compression, the power output is only about one and one half times that of the four-stroke cycle cylinder.

302. Fuels for Internal-combustion Engines.—The fuel entering the cylinder of an internal-combustion engine may be a gas, vapor, or a liquid in the form of a fine spray.

Liquid fuels include crude petroleum oils, petroleum distillates and alcohol. Crude oils are mechanical mixtures of hydrocarbon compounds, with varying amounts of impurities. The various hydrocarbons have different vaporization temperatures and can be separated from the crude oil by fractional distillation. In this process, the lighter distillates are vaporized at comparatively low temperatures and afterward condensed. At higher temperatures the heavier parts are vaporized and thus separated. After all the parts or fractions of the oil are driven off, paraffin or asphalt remains as a residue, depending upon the base of the crude oil. The distillates thus obtained include such liquids as gasoline, kerosene, lubricating oils and other compounds of hydrogen and carbon. An average yield of gasoline is 30 bbl. from 100 bbl. of crude oil. By special cracking processes (high pressure and temperature distillation), the heavier oils yield gasoline and naphtha, leaving coke as a residue.

Specific gravity of liquid fuels is ordinarily determined by a hydrometer reading, using the Baumé scale. The relation between degrees Baumé and specific gravity is:

$$\text{Sp. gr.} = \frac{140}{(130 + \text{deg. Bé.})} \qquad \text{liquids lighter than water}$$

$$\text{Sp. gr.} = \frac{145}{(145 - \text{deg. Bé.})} \qquad \text{liquids heavier than water}$$

The American Petroleum Institute uses the Baumé scale with slightly different constants:

$$\text{Sp. gr.} = \frac{141.5}{(131.5 + \text{deg. API})}$$

The characteristics of *gasoline* vary according to the crude oil and the method of refining. The specific gravity varies from 57 to 62°Bé., as an average. The analysis, by weight, is approximately as follows: carbon 84.76 per cent, hydrogen 15.24 per cent and the average lower heating value is 18,500 B.t.u. per pound.

Gasoline is a mixture of several more or less volatile hydrocarbons. However, octane (C_8H_{18}) is commonly thought of as gasoline and is used as a measure of certain gasoline properties. The so-called octane number, a system of measuring the knocking property of a gasoline, depends upon the knocking qualities of definite mixtures of octane (C_8H_{18}) and heptane (C_7H_{16}). Octane is practically free from knock while heptane knocks worse than any known gasoline. The octane number gives the percentage of octane in the mixture.

Kerosene is heavier than gasoline and has a specific gravity of about 45°Bé. The composition and heating value are approximately the same as for gasoline but the temperature required to vaporize it is considerably higher. Gas or "furnace oil" of 34 to 38°Bé. is being used in automotive oil engines such as truck, tractor, and aviation engines.

Fuel oil has properties similar to those of kerosene. It is, however, much heavier, has a specific gravity of approximately 26°Bé. and an average lower heating value of 19,000 B.t.u. per pound. This oil is cheap, safe to handle because of low volatility and is very desirable as an oil engine fuel.

Raw crude oils are generally not available, as a fuel. There is considerable variation in their composition, and this renders them unsatisfactory for most uses. An average value for the specific gravity of crude oil is 30°Bé. and the lower heating value varies from 18,000 to 21,500 B.t.u. per pound.

Alcohol has had little use as an engine fuel. It has, nevertheless, certain qualities that make its use practical in a mixture with other fuels. Ethyl alcohol (C_2H_6O) is less hazardous, in case of fire, than gasoline, and its vapor can be compressed to a higher pressure without danger of ignition. The specific gravity is approximately 45°Bé. and the lower heating value is about 12,000 B.t.u. per pound.

Gases used as engine fuels include producer gas, illuminating gas, coke-oven gas, blast-furnace gas, water gas and natural gas.

In the manufacture of *producer gas* (135 B.t.u. per cubic foot), coke, lignite or wood is subjected to a high temperature. Air containing water vapor is then passed through the hot bed of residual carbon. The gas formed by this process is composed of hydrocarbons, carbon monoxide, hydrogen, oxygen and carbon dioxide.

Illuminating gas is manufactured by distilling coal in sealed retorts. The volumetric analysis of this gas averages 46 per cent hydrogen, 37 per cent methane, 6 per cent carbon monoxide and small volumes of hydrocarbons, oxygen and carbon dioxide. The hydrocarbons form the illuminants. Coke remains as a residue. This gas is clean, high in heating value and desirable but very uneconomical to use as an engine fuel.

Coke-oven gas (400 to 500 B.t.u. per cubic foot) is a by-product from the manufacture of industrial coke which is produced by either the "beehive" oven or the more modern "by-product" oven. About half of the distilled gas is used for heating the ovens, and the excess gas is available for heating or power.

Blast-furnace gas is available in large quantities in localities where iron is manufactured and is often used as a fuel for blowing engines.

It is about one-fourth, by volume, carbon monoxide and has a heating value of approximately 100 B.t.u. per cubic foot. Small quantities of hydrogen and sometimes methane form the only other combustible constituents of the gas. Because of the dust and harmful metallic impurities, this gas must be carefully cleaned before being used as an engine fuel.

Natural gas (760 to 1,350 B.t.u. per cubic foot) is an excellent engine fuel where available. The supply is being rapidly depleted and there are few industrial cities that have this fuel available. Long-distance pipe lines are, however, changing this latter condition.

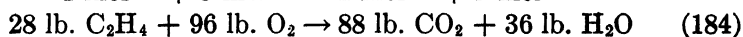
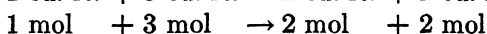
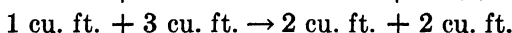
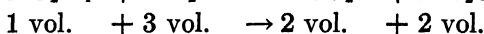
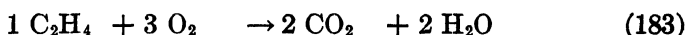
303. Combustion of Fuels in Internal-combustion Engines.—As in the case of fuels for the steam boiler furnace, carbon and hydrogen are the predominating heat-producing elements of fuels used in internal-combustion engines. Tables 3-4 and 3-5, page 57, include typical oil and gas fuels, showing the combinations of these elements, and listing their heating values. Table 3-7, page 65, includes useful combustion data of the common combustible elements and of widely used gaseous and vaporous fuels.

All of the principles discussed in Chap. III may be applied to combustion in an engine cylinder. However, combustion in an engine cylinder has at least two distinctive features. In the first place, it occurs in an extremely confined space, under conditions of high pressure and with very short time for its completion. Secondly, the fuel enters the combustion space as a gas, vapor or finely divided spray. Therefore a volumetric consideration of the fuel and air mixture and of the gaseous products of combustion is essential in the development of engine cylinders to attain high thermal efficiency and maximum power. If the volume in a cylinder at the end of compression (clearance volume) is made too small compared to the piston displacement, there is a probability that preignition, due to the high temperature of compression, will occur. Preignition of the fuel results in an increase in the work of compression, with a corresponding decrease in the net work of the cycle. On the other hand, if the ratio of clearance volume to piston displacement is made too large, the power output and thermal efficiency of the cycle are reduced.

304. Volumetric Relationships of Gases.—In Art. 44, page 61, are given combining chemical equations for certain combustible elements with oxygen, and it is pointed out that the combining equations indicate for the respective elements the proportions both by weight and by volume. The weight relation is obtained by substituting in the equation the molecular weights of the respective

elements, while the volume relation is indicated by the respective coefficients, which designate the number of units of volume.

Equation (183) gives the chemical equation for the perfect combustion of ethylene (C_2H_4) considered as a reaction of gases, and with the volumetric relation of the gases involved. The volumes may be expressed in any units of volumes, as for instance cubic feet or mols. The relation by weight may be determined by substituting in the equation the weight of each mol.

$$\begin{array}{lllll} 1\ C_2H_4 & +\ 3\ O_2 & \rightarrow 2\ CO_2 & +\ 2\ H_2O & (183) \\ 1 \text{ vol.} & +\ 3 \text{ vol.} & \rightarrow 2 \text{ vol.} & +\ 2 \text{ vol.} & \\ 1 \text{ cu. ft.} & +\ 3 \text{ cu. ft.} & \rightarrow 2 \text{ cu. ft.} & +\ 2 \text{ cu. ft.} & \\ 1 \text{ mol} & +\ 3 \text{ mol} & \rightarrow 2 \text{ mol} & +\ 2 \text{ mol} & \\ 28 \text{ lb. } C_2H_4 & +\ 96 \text{ lb. } O_2 & \rightarrow 88 \text{ lb. } CO_2 & +\ 36 \text{ lb. } H_2O & (184) \\ 1 \text{ lb.} & +\ 3.42 \text{ lb.} & \rightarrow 3.14 \text{ lb.} & +\ 1.28 \text{ lb.} & \end{array}$$

Considering the gaseous products resulting from combustion, nitrogen, inert but accompanying oxygen through the combustion process, must be included in the chemical equation. Again considering the combustion of ethylene:

$$1\ C_2H_4 + 3\ O_2 + 11.34\ N_2 \rightarrow 2\ CO_2 + 2\ H_2O + 11.34\ N_2 \qquad (185)$$

The coefficient of the nitrogen (N_2) is 3.78 times that of the oxygen (O_2) as explained in Art. 45, page 64. Equation (185) may be developed in units of volume (cubic feet or mols) or in units of weight (pound).

Compound gaseous fuels (see Table 3-5, page 58) are constituted of a mixture of different gases, of which part are combustible and part inert. The gas analysis, as determined, is on a volumetric basis, but, in calculations, it may be desirable to use the analysis by weight. This change may be effected by considering the unit volume of the compound gas to be 1 mol, then the volumetric analysis gives the part of a mol occupied by each constituent. For each gas, the weight is determined by the product of the decimal part of a mol by the weight per mol of the respective gas. This computation is carried out for each constituent, and the total of these results gives the total weight per mol of the compound gas. From this the analysis by weight is obtained.

In the study of combustion of compound gaseous fuels, three results are commonly ascertained, namely: the quantity of the gaseous products resulting from combustion, the quantity of air supplied for combustion, and the heating value of the fuel under consideration. If

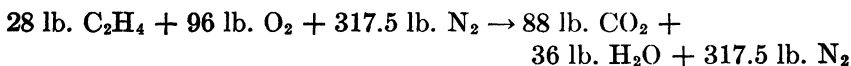
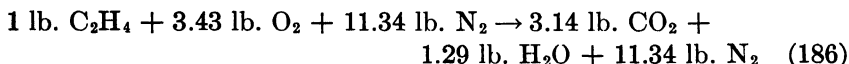
either the gaseous products of combustion, or the air supplied are desired on the basis of cubic feet per cubic foot of fuel, then a chemical equation for theoretical combustion must be set up. Such an equation is Eq. (185). Similar equations must be set up for each combustible constituent of the gas mixture. From Eq. (185), it may be seen that, for theoretical combustion of 1 cu. ft. of C_2H_4, 3 cu. ft. of O_2 and 11.34 cu. ft. of N_2, making a total of 14.34 cu. ft. of air, are required. If the combustion occurs with a certain percentage of excess air, assuming complete combustion, the volume of air supplied can be easily obtained. If, for example, 50 per cent excess air is used, the air supplied is 150 per cent of 14.34 or 21.51 cu. ft. per cubic foot of C_2H_4.

For the same gas, C_2H_4, for theoretical combustion, 1 cu. ft. C_2H_4 produces 2 cu. ft. CO_2, 2 cu. ft. H_2O and 11.34 cu. ft. N_2 as the gaseous products. It should be noted that H_2O is treated as a gas. If 50 per cent excess air is assumed, the volume of N_2 will be 11.34 cu. ft. of N_2 plus 50 per cent of 11.34, or 5.67 cu. ft. N_2, making a total of 17.01 cu. ft. of N_2. The O_2 of the excess air is determined from the excess N_2, hence the O_2 of the gaseous products is 5.67 cu. ft. divided by 3.78 or 1.5 cu. ft. of O_2 per cubic foot of C_2H_4.

The calculations on a weight basis are similar. Equation (185) is changed to a weight basis as follows:

$$28 \text{ lb. } C_2H_4 + 96 \text{ lb. } O_2 + 317.5 \text{ lb. } N_2 \rightarrow 88 \text{ lb. } CO_2 + 36 \text{ lb. } H_2O + 317.5 \text{ lb. } N_2$$

or

$$1 \text{ lb. } C_2H_4 + 3.43 \text{ lb. } O_2 + 11.34 \text{ lb. } N_2 \rightarrow 3.14 \text{ lb. } CO_2 + 1.29 \text{ lb. } H_2O + 11.34 \text{ lb. } N_2 \quad (186)$$

These values may be obtained from Table 3-7, page 65. From Eq. (186) it is seen that for theoretical combustion, 1 lb. C_2H_4 requires 14.77 lb. of air, and produces as gaseous products 3.14 lb. CO_2, 1.29 lb. H_2O and 11.34 lb. N_2. Assuming 50 per cent excess air, the nitrogen would be 11.34 lb. plus 5.67 lb., or 17.01 lb. N_2 per pound of C_2H_4. The oxygen would be 5.67 divided by 3.32 or 1.71 lb. O_2 per pound of C_2H_4.

To determine the volume of the gaseous products formed by the combustion of 1 lb. of a compound gas, the weight of each constituent in 1 mol of the compound gas is calculated. Then for each constituent, the weights of the gaseous products are determined for the combustion of 1 mol of the compound gas. From the gas law $PV_m = 1{,}544T$, the volume of 1 mol of the compound gas is calculated. Then by multiply-

ing the weights of the gaseous products from 1 mol by the volume of 1 mol of compound gas, the desired results are obtained.

To determine the weight of the gaseous products formed by the combustion of 1 cu. ft. of a compound gas, for each constituent, the cubic feet of the gaseous products per cubic foot of compound gas is first calculated. These results are divided by the respective specific volume (cubic feet per pound) of each constituent gas. The specific volume is determined from the volume per mol and the weight per mol.

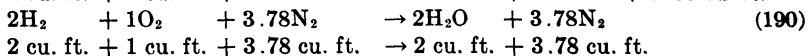
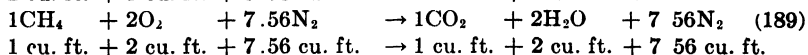
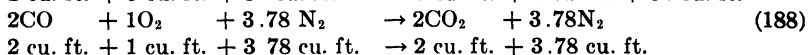
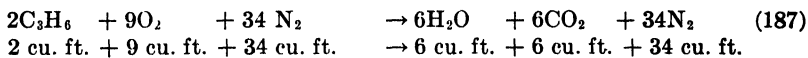
In determining the heating value of a hydrocarbon, the approximate method is by DuLong's equation (14,600C + 62,100H) in which C and H represent weight of carbon and hydrogen respectively as pound per pound of gas. To ascertain the heating value of a compound gas, as B.t.u. per cubic foot, the heating value is first calculated on a weight basis, as B.t.u. per mol of compound gas. This result is divided by the volume of 1 mol under the given conditions of pressure and temperature.

305. Typical Combustion Example.—The application of the principles of combustion to a gaseous fuel may be illustrated by the following solution of a typical example.

Example 15-2.—For the gas fuel, of which the analysis is included in the following data, determine the following values, each based on the theoretical requirement of air being supplied for combustion: (*a*) volume of air supplied per cubic foot of gas; (*b*) volume of the products of combustion per cubic foot of gas; (*c*) weight of air per pound of gas; (*d*) weight of products of combustion per pound of gas; (*e*) weight of products of combustion per cubic foot of gas at 50 lb. per square inch absolute and 100°F.; (*f*) heating value, B.t.u. per cubic foot at 50 lb. per square inch absolute and 100°F.

Gas analysis, percentage by volume: CO_2, 3.8; C_3H_6, 14.6; CO, 28.0; CH_4, 17.0; H_2, 35.6; N_2, 1.0.

Solution.—*a.* Volume of air required: The combining equations of the combustible constituents with oxygen are as follows:

$$\begin{array}{llllll}
2C_3H_6 & + 9O_2 & + 34\ N_2 & \rightarrow 6H_2O & + 6CO_2 & + 34N_2 \qquad (187) \\
2 \text{ cu. ft.} & + 9 \text{ cu. ft.} & + 34 \text{ cu. ft.} & \rightarrow 6 \text{ cu. ft.} & + 6 \text{ cu. ft.} & + 34 \text{ cu. ft.} \\
2CO & + 1O_2 & + 3.78\ N_2 & \rightarrow 2CO_2 & + 3.78N_2 & \qquad (188) \\
2 \text{ cu. ft.} & + 1 \text{ cu. ft.} & + 3.78 \text{ cu. ft.} & \rightarrow 2 \text{ cu. ft.} & + 3.78 \text{ cu. ft.} & \\
1CH_4 & + 2O_2 & + 7.56N_2 & \rightarrow 1CO_2 & + 2H_2O & + 7.56N_2 \qquad (189) \\
1 \text{ cu. ft.} & + 2 \text{ cu. ft.} & + 7.56 \text{ cu. ft.} & \rightarrow 1 \text{ cu. ft.} & + 2 \text{ cu. ft.} & + 7.56 \text{ cu. ft.} \\
2H_2 & + 1O_2 & + 3.78N_2 & \rightarrow 2H_2O & + 3.78N_2 & \qquad (190) \\
2 \text{ cu. ft.} & + 1 \text{ cu. ft.} & + 3.78 \text{ cu. ft.} & \rightarrow 2 \text{ cu. ft.} & + 3.78 \text{ cu. ft.} &
\end{array}$$

The air requirement in cubic feet per cubic foot of each constituent is obtained from the above equations.

1 cu. ft. C_3H_6 requires 4.5 cu. ft. O_2 or 21.5 cu. ft. air.
1 cu. ft. CO requires 0.5 cu. ft. O_2 or 2.39 cu. ft. air.
1 cu. ft. CH_4 requires 2 cu. ft. O_2 or 9.57 cu. ft. air.
1 cu. ft. H_2 requires 0.5 cu. ft. O_2 or 2.39 cu. ft. air.

The air required by 1 cu. ft. of gas is

0.146 cu. ft. $C_3H_6 \times 21.5$	= 3.14 cu. ft.
0.28 cu. ft. $CO \times 2.39$	= 0.67
0.17 cu. ft. $CH_4 \times 9.57$	= 1.63
0.356 cu. ft. $H_2 \times 2.39$	= 0.85
Volume of air per cubic foot gas	= 6.29 cu. ft.

b. Volume of the products of combustion: The gaseous products of combustion will be CO_2, H_2O and N_2 and their volumes are determined by use of the Eqs. (187) to (190) inclusive with the volumetric analysis of the fuel gas. Referring to Eq. (187), it is evident that the theoretical combustion of 1 cu. ft. C_3H_6 in air produces 3 cu. ft. H_2O, 3 cu. ft. CO_2 and 17 cu. ft. N_2. Since 1 cu. ft. of the fuel gas contains 0.146 cu. ft. of C_3H_6 the above results must be multiplied by 0.146. Similar calculations are carried out for each constituent and the results tabulated as follows:

PRODUCTS OF COMBUSTION, CUBIC FEET PER CUBIC FOOT OF GAS

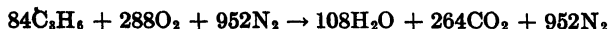
Gas	Cubic feet	CO_2	H_2O	N_2
CO_2	0.038	0.038		
C_3H_6	0.146	0.438	0.438	2.480
CO	0.280	0.280	.....	0.530
CH_4	0.170	0.170	0.340	1.285
H_2	0.356	.....	0.356	0.673
N_2	0.010	.....	.....	0.01
Total	1.00	0.926	1.134	4.978

The total volume of the products of combustion is the sum of the three constituents given in the above tabulation, or 7.038 cu. ft.

c. The volumetric analysis is changed to the weight basis as follows:

Gas	Part of 1 mol		Molecular weight		Pound per mol	Pound per pound
CO_2	0.038	×	44	=	1 673	0.0863
C_3H_6	0.146	×	42	=	6.141	0.3169
CO	0.280	×	28	=	7.843	0.4047
CH_4	0.170	×	16	=	2.725	0.1406
H_2	0.356	×	2	=	0.712	0.0370
N_2	0.010	×	28	=	0.280	0.0145
Total	1.000				19.374	1.0000

Substituting the molecular weights in the combustion Eq. (187), the following equation is obtained:

$$84C_3H_6 + 288O_2 + 952N_2 \rightarrow 108H_2O + 264CO_2 + 952N_2$$

Hence

1 lb. C_3H_6 requires 3.43 lb. O_2 or 14.82 lb. air.

Similarly:

1 lb. CO requires 0.572 lb. O_2 or 2.46 lb. air.
1 lb. CH_4 requires 4 lb. O_2 or 17.28 lb. air.
1 lb. H_2 requires 8 lb. O_2 or 34.56 lb. air.

Hence, the theoretical weight of air to burn 1 lb. of the fuel can be calculated:

C_3H_6 .... 0.3169 lb. × 14.82 = 4.70 lb.
CO .... 0.4047 lb. × 2.46 = 1.00
CH_4 .... 0.1406 lb. × 17.28 = 2.43
H_2 .... 0.0370 lb. × 34.56 = 1.27
Pound of air per pound of fuel total = 9.40 lb.

d. The weight of the gaseous products of combustion are obtained by using Eqs. (187) to (190) with the molecular weights substituted. From the equation in *c*, above, it is evident that 1 lb. C_3H_6 produces 1.285 lb. H_2O, 3.14 lb. CO_2 and 11.32 lb. N_2. Hence the burning of 0.3169 lb. C_3H_6 would produce 0.408 lb. H_2O, 0.996 lb. CO_2 and 3.60 lb. N_2. The remaining combustible constituents are handled the same way and the results are listed in the following tabulation.

PRODUCTS OF COMBUSTION, POUNDS

Gas	Part of 1 lb.	CO_2	H_2O	N_2
CO_2	0.0863	0.0863		
C_3H_6	0.3169	0.996	0.408	3.60
CO	0.4047	0.636	......	0.768
CH_4	0.1406	0.386	0.316	1.862
H_2	0.0370	......	0.33	0.972
N_2	0.0145	......	......	0.0145
Total	1.0000	2.1043	1.054	7.2165

Total products of combustion, 10.3748 lb. per pound of fuel.

e. Weight of products of combustion per cubic foot of gas fuel at 50 lb. per square inch and 100°F.

From section *c* and *d* and Table 3-7:

Gas	Pound per mol	Gaseous products, lb. per mol		
		CO_2	H_2O	N_2
CO_2	1.673	1.67		
C_3H_6	6.141	19.25	7.88	69.6
CO	7.843	12.3	.....	14.8
CH_4	2.725	7.5	6.13	36.2
H_2	0.712	.....	6.41	18.9
N_2	0.280	.....	.....	0.28
Total	19.374	40.72	20.42	139.78

Volume of 1 mol at 50 lb. per square inch and 100°F.:

$$V_m = \frac{1{,}544T}{P} = \frac{1{,}544 \times 560}{50 \times 144} = 120 \text{ cu. ft.}$$

Gaseous products, pounds per cubic foot of gas fuel:

CO_2	40.72 lb. ÷ 120	= 0.339
H_2O	20.42 lb. ÷ 120	= 0.171
N_2	139.78 lb. ÷ 120	= 1.163
Total		1.673

f. Heating value of 1 cu. ft. at 50 lb. per square inch and 100°F. From section *e* and Table 3-7:

Gas	Pound per mol	B.t.u. per lb. of constituent	B.t.u. per mol of fuel gas
CO_2	1.673		
C_3H_6	6 141	× 21,390	= 131,000
CO	7 843	× 4,380	= 34,300
CH_4	2 725	× 23,850	= 65,100
H_2	0 712	× 62,100	= 44,200
N_2	0.280		
Total	19.374		274,600

From section *e*, volume of 1 mol = 120 cu. ft.

$$\text{B.t.u. per cubic foot at 50 lb. and 100°F.} = \frac{274{,}600}{120} = 2{,}288$$

Heating value of C_3H_6 (DuLong's formula):

$$HV = 0.857 \times 14{,}600 + 0.143 \times 62{,}100 = 21{,}390 \text{ B.t.u. per pound}$$

306. Development of Internal-combustion Engines.—The internal-combustion engine has been developed extensively for furnishing power for both transportation and stationary purposes. For transportation, it greatly outnumbers other prime movers, and, for aviation, it has made great strides in development and refinement, especially during the period since the World War. For railway locomotives, the steam engine predominates, but a beginning has been made in the use of oil engines for this purpose. The trend of automotive engine development has been toward a continual increase in the size of unit and the ability to run constantly at high speeds. The characteristics which are stressed in the design of automobile engines are power, speed, beauty and reliability, but not necessarily fuel economy. All of these merely follow the demands of the buying public. Aviation engine

development has followed similar lines, stressing power and reliability rather than economy. The use of the oil engine for this purpose has as its aim increased safety by removing the fire hazard present when gasoline is used as a fuel.

In the development of the internal-combustion engine for stationary power, economy of operation is always emphasized, chiefly because of its competition with steam plants. Gas engines for this purpose generally use cheap fuels, such as by-product, blast-furnace gas, etc. Oil-engine builders have always featured the low quantity

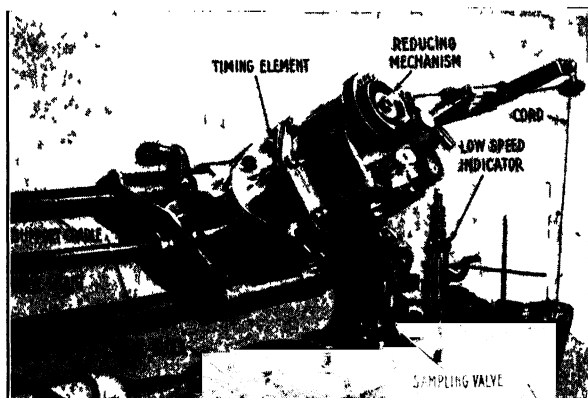



FIG. 303.—Jacklin high-speed indicator.

of fuel required to produce a horsepower-hour as a selling point. Other refinements in internal-combustion engines have included a reduction in weight, increased speed and improvement in the combustion equipment.

307. Performance.—Internal-combustion engines may be tested for the purpose of determining the brake horsepower, indicated horsepower, fuel economy, thermal efficiency, thermal heat balance, etc. For speeds of 300 r.p.m. or less, an engine indicator similar to that used for steam engines can be used in determining the mean effective pressure of the engine. Above this speed, diagrams from such an indicator are inaccurate, due to the inertia of the pencil arm and other moving parts. At speeds as low as 300 r.p.m. one complete cycle may require but $\frac{1}{5}$ sec., which is a short time allowance in which the indicator mechanism is to operate.

308. High-speed Indicator.—Various indicators have been devised for obtaining indicator diagrams from engines operating at high speeds. Of these is the *Jacklin high-speed indicator* shown in Fig. 303. This indicator has been constructed so as to obtain representative

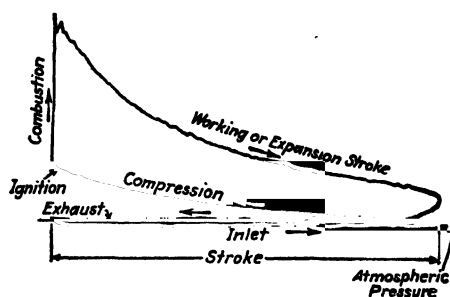
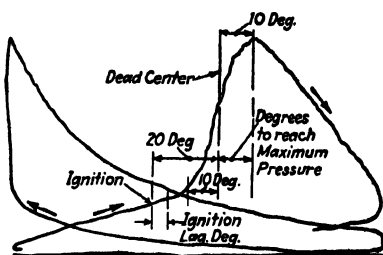
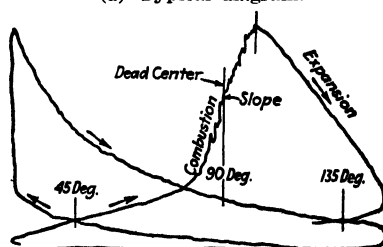
diagrams from engines running as fast as 4,000 r.p.m. The essential mechanism includes the ordinary type of low-speed indicator. Between this indicator and the engine cylinder is a small mechanically operated poppet valve which is opened for a short interval of time in each cycle. The opening permits a small amount of gas to pass into or from the indicator cylinder so as to bring about the same

(*a*) Typical diagram.

(*b*) Regular and off-set diagrams.

Fig. 304.—Indicator diagrams taken with Jacklin high-speed indicator.

pressure in the indicator cylinder as is exerted in the engine cylinder at that instant of the cycle. Fourteen hundred or more cycles of a four-stroke engine, and eight hundred or more cycles of a two-stroke engine, are required to obtain a complete diagram. The drum of the indicator is moved by a cord attached to the timing mechanism. The time of opening of the poppet valve is changed by a small amount each cycle, in harmony with the drum motion, so that the pressure in the

indicator cylinder registers correctly for the corresponding position of the indicator pencil. A typical diagram taken with this indicator is shown in Fig. 304 *a*.

On using this indicator for investigating combustion processes and valve functioning, it is of advantage to obtain offset diagrams in which the true pressures are registered with the indicator pencil 75 to 90 deg. out of phase with the engine piston. Figure 304 *b* shows the regular diagrams together with the offset diagrams, which have been taken with the pencil set 75 deg. out of phase with the engine piston. This change is effected by a suitable adjustment in the timing mechanism of the indicator. The result of the change is to bring ignition and combustion near the center of the diagram where the

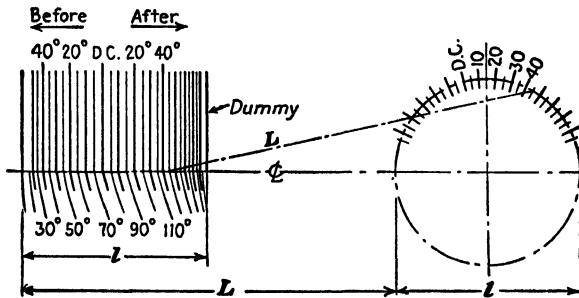



Fig. 305.—Crank-angle dummy drawing.

speed of the indicator drum is maximum. In this way, the combustion process is spread out and its effect is clearly seen.

Crank-angle positions, before and after dead center, can be laid off on the diagrams of this type by the use of a crank-angle dummy to be explained later. On the offset diagram, the ignition lag and other peculiarities of combustion can be inspected. Ignition on the offset diagrams of Fig. 304 *b* occurs at 20 deg. before dead center, and it may be seen that there is no particular increase in pressure until 10 deg. before dead center. The pressure rises rapidly from this point until the maximum is reached at 10 deg. after dead center. The combustion requires an appreciable time before expansion begins.

To use the offset diagram accurately, a so-called *crank-angle dummy* drawing, as shown in Fig. 305, is made. A circle is drawn, whose diameter is equal to the length of the indicator diagram, and 5-deg. crank-angle points located thereon. Dividers are set to a length proportional to the length of the engine connecting rod. One point of the dividers is placed on the 5-deg. points, successively, and arcs are struck across the horizontal center line. Vertical lines are then drawn at these points, and the corresponding crank-angle positions

are identified. The dummy drawing is then placed over an illuminated glass and the offset diagram oriented over it, so that the atmospheric line of the diagram coincides with the horizontal center line of the drawing. The ends of the diagram and the dummy should also coincide. The crank-angle positions are then traced on the offset diagram.

309. Indicated Horsepower.—The indicated horsepower of an internal-combustion engine may be calculated from regular indicator diagrams. Thus

$$\text{i.hp.} = \frac{PLAN_P}{33{,}000} \times n \qquad (191)$$

in which

P = mean effective pressure taken from the indicator diagram, lb. per square inch.
L = length of engine stroke, ft.
A = net piston area, sq. in.
N_P = number of power strokes per minute, per cylinder.
n = number of cylinders.

Commonly the engine is single acting, in which case the total piston area is effective. For a double-acting engine the indicated horsepower of the head end is added to the indicated horsepower of the crank end to obtain the total indicated horsepower of each cylinder. In the latter case the values of P and A are almost always different for the two ends of the cylinder.

It is difficult to obtain an accurate value of the indicated horsepower of an internal-combustion engine because of the extreme difficulty in ascertaining an accurate, representative value of the mean effective pressure or pressure-volume diagram of the cycle. This is due in part to the high speeds common to internal-combustion engines, the errors due to inertia of the moving parts of the common indicating mechanism, and in part to the wide variation in the pressures attained in successive cycles. In testing, indicator diagrams are taken as frequently as practicable.

310. Brake Horsepower.—The brake horsepower may be ascertained by means of a Prony brake, electric dynamometer, water brake, or fan dynamometer. The Prony brake is described under steam-engine testing.

The Sprague electric dynamometer (Fig. 306) may be used either as a dynamo for power absorption or as a motor for measuring friction losses. The power shaft of the prime mover under test is coupled to the armature shaft. The floating field magnets are supported on

ball bearings so that the frame with contained field magnets has free rotation through a small arc. The torque produced by rotating the armature in the magnetic field of the field magnets tends to make the frame rotate with the armature. This rotation is restrained by scales or other balancing mechanism. When the armature is rotating, the reaction torque, which is equal to and opposite to the driving torque, may be measured by the scales. The calculation of the brake horsepower of the connected prime mover is exactly the same as with the Prony brake.

The variation in load on the prime mover tested is accomplished by a dual control of current strength in the armature and field windings. This dual control permits speed adjustment from zero to maxi-

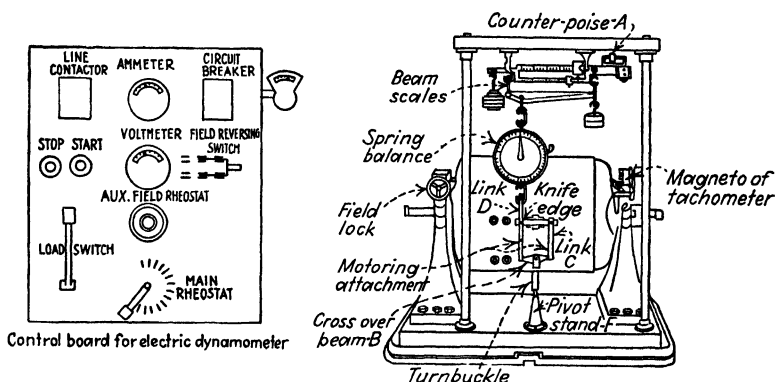



FIG. 306.—Sprague electric dynamometer.

mum and torque adjustment to any value between minimum and maximum.

Direct current is used because of more accurate control obtained, as compared with alternating current.

The Alden brake is a fluid-friction brake capable of measuring power of prime movers including those of large size. A smooth cast-iron disc is keyed on the rotating shaft of the prime mover. This is enclosed in a cast-iron shell which is free to revolve about the shaft irrespective of the shaft rotation. A flexible copper disc is fitted to the interior of each side of the shell enclosing between the discs and the shell water-tight spaces through which water flows, maintaining a uniform temperature. The small clearance space between the two copper discs, and in which the cast-iron disc rotates, is filled with oil. Increasing the water pressure forces the two copper discs inward against the rotating cast-iron disc. The resulting friction tends to rotate the shell in the rotational direction of the shaft. The turning

effect is measured by a radial arm which is fixed to the cast-iron shell, and the brake horsepower is calculated just the same as when using the Prony brake.

The following formula for brake horsepower is used with the Prony brake, electric dynamometer, and water brake:

$$\text{b.hp.} = \frac{2\pi RwN}{33{,}000} \tag{192}$$

in which

R = length of brake radius, ft.
w = net weight required to balance the force of the arm, lb.
N = r.p.m.

If

$$K = \frac{33{,}000}{2\pi R} = \text{constant of the testing machine,}$$

$$\text{b.hp.} = \frac{wN}{K} \tag{193}$$

311. Horsepower Rating Formula.—The commonly used horsepower rating formula was originated by the old American League of Automobile Manufacturers (A.L.A.M.). It is based on an average value of piston speed of 1,000 ft. per minute and a brake mean effective pressure of 67.5 lb. per square inch. This formula is as follows:

$$\text{b.hp.} = \frac{nb^2}{2.5} \tag{194}$$

in which

n = number of cylinders.
b = bore or diameter of cylinder, in.

312. The Brake Mean Effective Pressure.—The mean effective pressure, which, if acting on the piston, would develop power equivalent to the brake horsepower is termed the brake mean effective pressure. Hence,

$$\text{b.hp.} = \frac{P_B LAN_P}{33{,}000} \times n$$

or

$$P_B = \frac{\text{b.hp.} \times 33{,}000}{LAN_P n} \tag{195}$$

in which

P_B = brake mean effective pressure.
L = length of piston stroke, ft.

A' = net piston area, sq. in.
N_P = number of power strokes per minute, per cylinder.
n = number of cylinders.

The brake mean effective pressure affords an accurate means of comparison of engine performance, and is used as such in preference to the indicated mean effective pressure. This is due to the inherent inability of the ordinary indicator to give a true diagram. Because of the variable factors that influence combustion, such as the amount of burned gas left in the cylinder, the air-fuel ratio, etc., the successive cycles of an internal-combustion engine vary to a greater or lesser degree. This variation is eliminated, however, in the mean effective pressure which is based on the brake horsepower.

Example 15-3.—Calculate the brake mean effective pressure for the Franklin six-cylinder, four-cycle, automobile engine, from the following data: cylinder $3\frac{1}{2}$ by $4\frac{3}{4}$ in.; 52 b.hp.; speed, 1,500 r.p.m.

Solution.—From Eq. (195),

$$\text{brake m.e.p.} = \frac{52 \times 33{,}000 \times 12 \times 2}{4.75 \times 9.64 \times 1{,}500 \times 6} = 100$$

The brake mean effective pressure of the Packard nine-cylinder, aviation oil engine, illustrated in Fig. 326, page 514, varies from 114 to 120 lb. per square in.

313. Torque.—The torque of an engine is the turning moment on the pounds of force exerted at a radius of 1 ft. Torque may be expressed as follows:

$$T = \frac{\text{b.hp.} \times 33{,}000}{2\pi N} = 5{,}252\,\frac{\text{b.hp.}}{N} = Rw \qquad (196)$$

in which

T = torque at 1 ft. radius, lb.
N = r.p.m. of engine.
R = radius of brake arm, ft.
w = net weight to balance force on arm, lb.

Example 15-4.—Determine the torque exerted by the Franklin engine for the data given in Example 15-3.

Solution:

$$T = 5{,}252\frac{52}{1{,}500} = 182 \text{ lb.}$$

314. Thermal Efficiency.—Thermal efficiency is the ratio of the output of the engine to the input. The output is the heat equivalent of 1 hp.-hr. (2,545 B.t.u.), and may be based on either the brake or indicated horsepower. Therefore, the thermal efficiency may be

based on either horsepower, and the one used should be spécified. The input, in B.t.u., is obtained from the product of the weight of fuel (pound) per horsepower-hour and the heating value (B.t.u. per pound of fuel). For gaseous fuels, the heat input is based on the volume instead of the weight of the fuel. Thermal efficiency may be expressed as follows:

$$e_t = \frac{2{,}545}{w_f \times \text{B.t.u. per pound}} \tag{197}$$

in which

w_f = weight of fuel per horsepower per hour, lb.

The volume of gas fuels used is measured under actual pressure and temperature conditions, while the heating value per unit volume is determined under standard conditions. These conditions, according to the A.S.M.E. Test Code, are 14.7 lb. per square inch absolute pressure, and 68°F. temperature. The perfect gas law [Eq. (4), Chap. II] is used to change the measured volume to standard conditions. Thus.

$$\frac{14.7 \times 144 \times V_o}{528} = \frac{P_1V_1}{T_1}$$

$$V_o = 0.25\left(\frac{P_1V_1}{T_1}\right) \tag{198}$$

In Eq. (198), the subscripts o and 1 refer to standard and actual conditions, respectively.

315. Volumetric Efficiency.—Volumetric efficiency is the ratio of the volume of the air or fuel mixture drawn into the cylinder during the suction stroke to the piston displacement per stroke. In either case, the volume of the charge must be corrected to standard conditions. The relative volumetric efficiency of a four-stroke cycle engine can be determined from the lower part of an indicator diagram taken using an indicator spring of low value. On taking the diagram for this purpose, excessive rise of the indicator pencil can be prevented by placing a sleeve around the spring to limit its compression. This magnifies, vertically, the lower part of the ordinary diagram so that the relation of the pressure of the atmosphere, exhaust and intake can be clearly seen.

FIG. 307.—Illustrating method of determining volumetric efficiency by the use of indicator diagrams.

Such a diagram is shown in Fig. 307. As shown, b is the distance between the intersection of the atmosphere line with, first, the suction

line and, second, the compression line, and a is the length of the diagram. The volume corresponding to b represents the volume of the charge, at atmospheric pressure, drawn into the cylinder per stroke. For accurate work this should be corrected for standard temperature. With these measurements, in inches, the volumetric efficiency e_v in percentage may be calculated as follows:

$$e_v = \frac{v}{a} \times 100 \tag{199}$$

Approximate values of volumetric efficiency are as follows: slow-speed, high-compression oil engines 93 per cent; throttling-governed gas engines 85 per cent; high-speed, low-compression carburetor engines 65 per cent.

316. Air-fuel Ratio.—The air-fuel ratio is expressed as the pounds of air supplied to an engine for each pound of fuel. Theoretically the air-fuel ratio may be determined from the combustion equations for the fuel, assuming perfect combustion. In actual combustion there is wide variation from the theoretical values. The following table gives the limits of explosiveness of common fuels.

TABLE 15-1.—EXPLOSIVE LIMITS

Fuel	Air-fuel ratio: Theoretical	Exploding: Min.	Exploding: Max.	Fuel	Air-fuel ratio: Theoretical	Exploding: Min.	Exploding: Max.
Hydrogen	34.56	5	340	Ethane	16.13	7	30
Carbon monoxide	2.46	0.3	7.3	Benzene	13.3	4	26
Methane	17.28	11	33	Gasoline	15.19	4	18
Acetylene	13.29	1	43	Methyl alcohol	6.5	2	14
Ethylene	14.81	4	30	Ethyl alcohol	9.0	3	18

The desired air-fuel ratios do not range from one limit of inflammability to the other, but range from those mixtures resulting in maximum economy to those of maximum power. Figure 308 shows curves plotted for a typical fuel at a certain speed. Using relative load (1.0 being full load) as one ordinate and air-fuel ratio as the other, curves for maximum power and maximum economy and also for the lean and rich limits for explosiveness have been plotted. Maximum power is most desirable near full load for any speed. Hence the most useful mixtures would follow the curve giving maximum economy

up to about 0.7 of full load, and then curve down to maximum power curve at full load.

On test, consumption of gaseous fuels is measured by a gas meter, or by orifice meters. Liquid fuels such as gasoline, fuel oil and kerosene are generally measured by actual weighing, although volumetric measuring devices are sometimes used. Air measurement is accomplished by an orifice meter or low-pressure nozzle, as described under Air flow, page 234. An indirect method of determining the weight of air used is from an Orsat analysis of the exhaust gases.

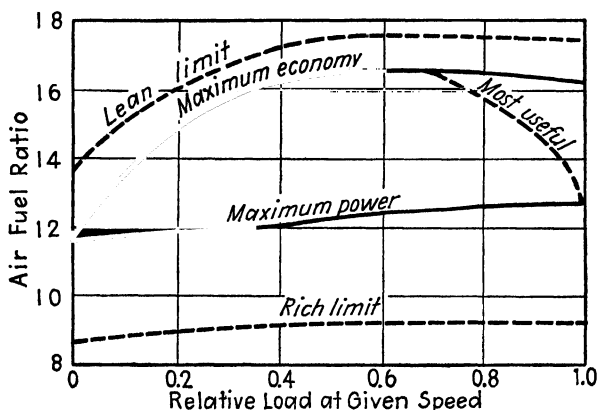



FIG. 308.—Air-fuel ratio, practical limits.

317. Testing.—Gasoline engines are tested according to the rules of the Society of Automotive Engineers. S.A.E. testing forms for recording such tests, composed of four sheets, are as follows:

1. Rules and directions.
2. Specification sheet.
3. Log sheet.
4. Curve sheets.

A brief résumé of the directions follows, with engine tests and curves. A complete test (Table 15-2) includes the determination at different speeds of:

1. Maximum horsepower.
2. Fuel economy at maximum horsepower at ¾, ½, and ¼ maximum horsepower at each speed.
3. Friction horsepower.

From these results the following curves are plotted (Fig. 315*b*, page 498):

1. Torque, r.p.m.
2. Maximum horsepower, r.p.m.
3. Brake mean effective pressure, r.p.m.

4. Friction horsepower, r.p.m.
5. Mechanical efficiency, r.p.m.
6. Fuel per brake horsepower per hour at different loads and speeds, lb.
7. Thermal efficiency at different loads and speeds, per cent.

For horsepower and fuel-economy tests it is recommended that runs be made at speed intervals of approximately 200 r.p.m. The duration of the brake-horsepower tests should not be less than 3 min. Where fuel consumption is measured, the duration of tests should not be less than 5 min. The duration of friction-horsepower tests should not be less than 2 min.

Before taking brake readings the dynamometer should be properly balanced. For the electric-cradle type this balancing is accomplished as follows:

The dynamometer is run idle as a motor, and a suitable counterbalance on the field frame is adjusted so that the scales read zero. This reading should be obtained with the dynamometer rotating first in one direction and then the other. The reaction of the armature on the field frame will exactly balance the friction of the brushes and armature bearings carried in the field frame. With the armature still rotating, check weights should be hung from the knife edge on the dynamometer arm. If the reading recorded is equal to the known weight applied, the dynamometer is balanced.

The method for measuring fuel consumption is by recording the decrease in weight of a tank from which fuel is being fed to the carburetor. In order to afford a fixed basis of comparison it is recommended that the outlet-water temperature for engines be kept at 175°F. (plus or minus 5°). Control is accomplished by a thermostat or by external control of quantity or temperature.

The approximate friction horsepower of an engine can be measured by means of the electric-cradle dynamometer. The dynamometer is used to drive the engine at various speeds, and the torque reaction is measured. This will be in the opposite direction to that obtained while the engine is driving the dynamometer, so that provision must be made for measuring the torque on both sides. The friction-horsepower test should be made immediately after the brake-horsepower test and before the engine has cooled. Approximate indicated horsepower is obtained by adding to the brake horsepower at any given speed, the friction horsepower obtained at the same speed.

318. Automotive Engines.—Present-day engines of this type are the result of many years of extensive engineering research and development. In general, they are quite similar, but each different make has so many features incident to its particular design that a detailed analysis of automotive engines cannot be made here.

TABLE 15-2.—TESTS OF 1928 CHEVROLET ENGINE

S.A.E. GASOLINE ENGINE TESTING FORMS **LOG SHEET—C NO.**

Name: Chevrolet
Model: 1928 No. Cyls. 4
Bore: 3 11/16 in. Stroke 4 in. Displ. (D) 170.8 cu. in.
Laboratory: Date: 12/20/35
Observers:

Fuel B.t.u. per lb. 18,500. Sp. gr. at °F.
Dynamometer: Diehl Arm (R): ft.
Humidity: per cent.
Oil: Grade: Cold test °F.
Saybolt univ. vis. at 130°F. at 210°F.

Run number	Symbol	Formula	1½ load	4/4 load	¾ load	½ load	¼ load
Brake horsepower and fuel consumption							
Time started		*					
Duration of run—min.	t	*	1.51	1.38	1.73	1.85	2.66
Counterstart	C_o	*	0	0	0	0	0
Counterfinish	C_t	*	2,923	1,481	1,765	1,984	2,907
Total revolutions	r	$C_t - C_o$	2,923	1,481	1,765	1,984	2,907
Average r.p.m.	N	$\frac{r}{t}$	1,935	1,072	1,020	1,070	1,090
Barometer, in. mercury		*	29.51	29.51	29.51	29.51	29.51
Room temp.		*	68	68	68	68	68
Correction factor	C.F.	$\frac{P_s}{P_o} \times \sqrt{\frac{T_o}{T_s}}$	1.02	1.02	1.02	1.02	1.02
Brake load at arm R	P	*	64.4	72.3	54	36 7	19.3
Brake load corrected	P_c	$CF \times P$	65.6	73 7	55.1	37 4	19.7
Torque, lb.-ft.	T	PR	84.5	95	70.9	47.1	25.3
Brake m.e.p.	np	$\frac{150.8\ T}{D}$	75	84.4	62.9	41.8	22.5
Brake hp.	b.hp.	$\frac{PRN}{5{,}252.1}$	31.5	19.4	13.78	9.8	5.25
Friction hp. at N	f.hp.	f.hp. curve	4.8	2 21	2 41	2.68	2.7
Indicated hp.	i.hp.	b.hp. + f.hp.	36 3	21.61	16 19	12.48	7.95
Mechanical efficiency	m.e.	$\frac{\text{b.hp.}}{\text{i.hp.}}$	0.868	0.892	0.85	0.793	0.66[illegible]
Temp. jacket water—in		*					
Temp. jacket water—out		*	154	164	150	152	150
Temp. oil—in		*					
Temp. oil—out		*					
Oil pressure, lb.		*					
Temp. air to carburetor		*					
Weight fuel start	W_o	*					
Weight fuel finish	W_t	*					
Lb. fuel used	W	$W_o - W_t$	0 5	0.25	0.25	0.25	0.25
Lb. fuel per b.hp.-hr.	F	$\frac{60W}{t \times \text{b.hp.}}$	0.632	0.562	0.63	0.828	1.07[illegible]
Thermal efficiency re. b.hp.	T.E.	$\frac{2{,}545}{F \times \text{B.t.u.}}$	21.8	24.5	21.8	16.65	12.8
Manifold pressure			14.1	13 2	18.5	22.4	25.7
Manifold pressure			12.4	12.5	11.7	11	10.5
Friction hp.							
Time started		*					
Duration, of run—min.	t	*					
Counterstart	C_o	*					
Counterfinish	C_t	*					
Average r.p.m.	n	$\frac{C_t - C_r}{t}$	1,850	1,025	980	1,020	1,030
Brake load at arm R	p	*	10.4	8.65	9.9	10 5	10.5
Friction hp.	f.hp.	$\frac{pRn}{5252.1}$	4.8	2.21	2.41	2.68	2.7
Mean temp. jacket water		*					

Above computed data corrected for barometer Yes and temperature Yes
No No

Refer to Specification Sheet No.
Refer to Curve Sheet No.

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Automobile engines for stock cars are built with 4, 6, 8, 12 and 16 cylinders, and more or less experimental work is being done to develop automobile engines with a greater number of cylinders. Among the many changes that are continually taking place in the design of automobiles is the front-wheel drive. Several manufacturers are now producing cars thus equipped. Most automobiles are equipped with water-cooled engines. There are, however, several cases in which the engine is designed for air cooling.

319. The Franklin Air-cooled Automobile Engine.—One of the early American automobiles is the Franklin automobile, cooled by air flow around the cylinders (Fig. 309). The cooling air, on flowing

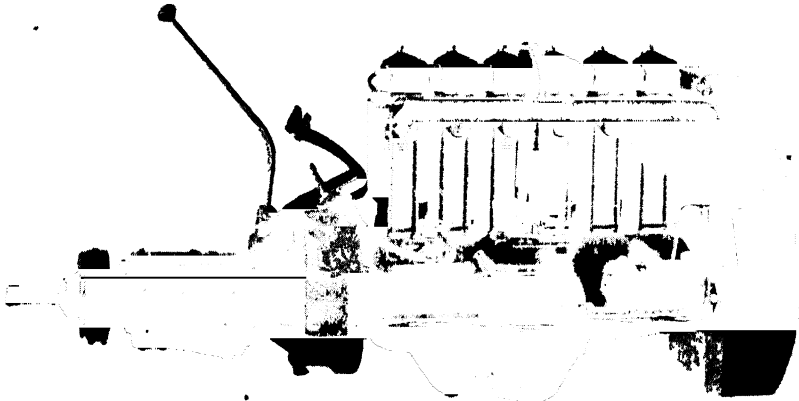



Fig. 309.—Franklin six-cylinder automotive engine, 1930 model.

through the flanges of the cylinders, takes a horizontal course from one side of the cylinder to the other. In the former models vertical fins were employed. The cylinders are 3½ in. in diameter and the piston stroke is 4¾ in.

The complete cooling system includes the fan, fan housing, cylinder barrel and cylinder head. The fan housing is shaped so that the air, free from eddy currents, enters at the front of the engine and flows through the cylinder housing along the sides of the cylinders. The housing is without deflectors, and the air is proportioned to keep all cylinders at the same temperature. In previous models, the vertical fins along the cylinder barrel had to effect the cooling of the cylinder head also. In this later design, the cylinder head has its own cooling flanges. The flow of air is directed so that only approximately 28 per cent passes over the cylinder barrel and the rest over the heads. The head is made from an aluminum casting, in order to obtain the advan-

tage of high heat conductivity, and the barrel is of cast iron. The cooling area of the head is 495 sq. in. and the cylinder barrel 590 sq. in., giving a total cooling area of 1,085 sq. in.

The valves are on the opposite sides of the cylinder, the intake valves of this engine being on the side of the cylinder where the cooling air enters the cooling flanges. This has a cooling effect on the fuel and air mixture entering the cylinder, and permits increasing the density of the charge, which in turn increases the mean effective pressure in the engine cylinders.

A multiblade, centrifugal fan is used in the air-cooling system and it is placed on the crank shaft, at the front of the engine. The fan is

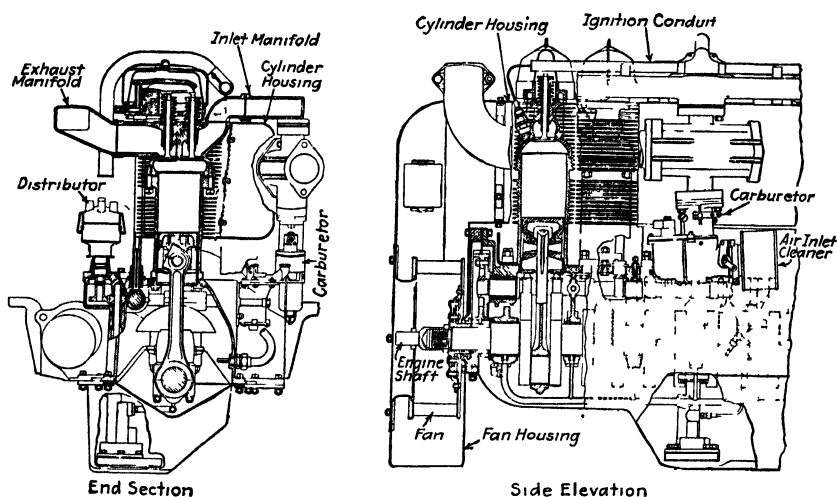



FIG. 310.—Sectional views of Franklin engine.

12 in. in diameter, and, at an engine speed of 3,000 r.p.m. (60 miles per hour), it delivers 62.5 cu. ft. of air per second against a pressure of 3.5 in. of water. Under these conditions it absorbs 8 hp. from the engine.

The use of large intake and exhaust valves reduces the throttling effect on the inlet and exhaust gases flowing through the ports.

The engine is continually supplied with lubricating oil which is circulated by a gear pump from the bottom of the crank case. The pump delivers the oil to a header from which it is distributed to the various bearings. A relief valve between the pump and header prevents a pressure rise above 50 lb. per square inch, and an oil purifier is used to maintain the lubricant in a clean condition. Sectional views of the Franklin engine are shown in Fig. 310.

A 19-plate storage battery furnishes current for the electrical circuit which includes a gear-driven generator. A thermostatic control is provided on the generator for decreasing the charging rate when the engine becomes hot. This prevents overheating and overcharging of the battery on long trips.

The power and torque performance curves for the Franklin engine (series 145) are shown in Fig. 311. It may be noted that a maximum of 95 hp. is obtained at 3,100 r.p.m.

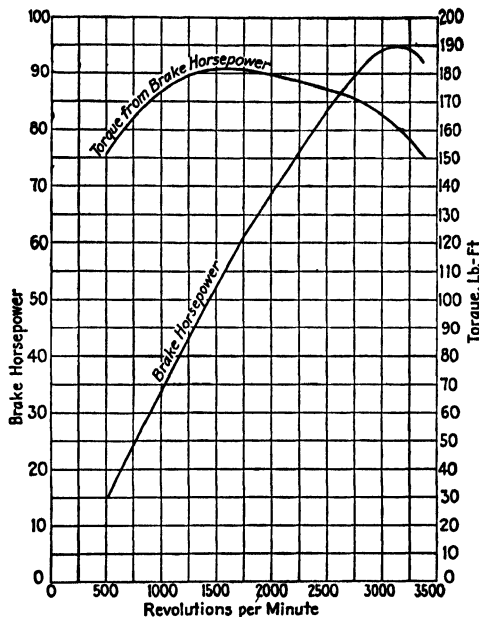



FIG. 311.—Horsepower and torque curves for Franklin six-cylinder engine.

The four-speed transmission, generally used on automobile engines, has speed ratios of 3.494, 1.993, 1.278 and 1:1.

320. The Studebaker, Water-cooled Automobile Engine.—Many of the *water-cooled engines* used for automotive purposes are of the eight-cylinder design. Eight-cylinder engines are built with all cylinders in line, or with two rows of four, forming a V angle, symmetrical about the vertical center line. However, the most popular engine is the six-cylinder model. The Studebaker engine, Dictator model, illustrated in Fig. 312, is typical of the latter style of design. Sectional views are shown in Figs. 313 and 314.

The cylinders are of cast iron, cast *en bloc*, that is, all contained in one casting. The assembled engine consists of three castings: the cylinder heads, the cylinder block and the lower part of the crank

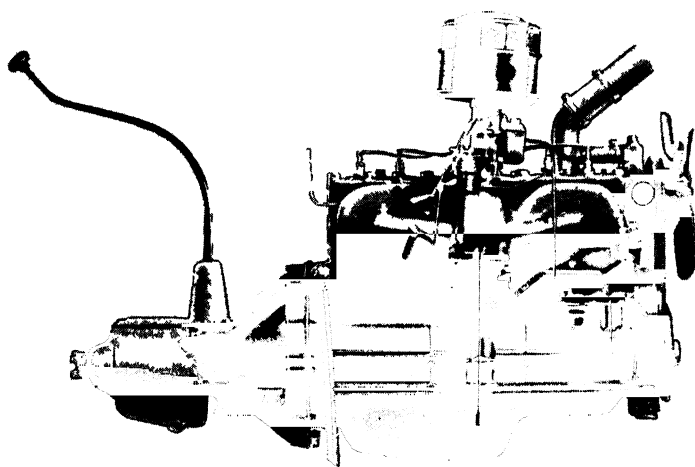



FIG 312—Studebaker six-cylinder Dictator engine.

EXHAUST MANIFOLD
CYLINDER HEAD
INLET MANIFOLD
VALVE COVER PLATE
CRANKSHAFT
CRANKCASE VENTILATOR
REAR CAMSHAFT BUSHING
AUTOMATIC CHOKE THERMOSTAT
CARBURETOR ASSEMBLY
INTAKE MANIFOLD HEATER VALVE ASSEMBLY
PISTON
PISTON RINGS
PISTON PIN
PISTON PIN CLAMP SCREW
WATER OUTLET
WATER BY PASS
HEATER CONNECTION
FAN
WATER PUMP
GASKET
TIMING GEAR COVER
CAMSHAFT GEAR
CAMSHAFT
CAMSHAFT FRONT BEARING
GEAR RETAINING SCREW
TIMING POINTER (SEE VIBRATION DAMPER ILLUSTRATION)
VIBRATION DAMPER
OIL RETAINING WASHER
OIL THROW RING
CRANKSHAFT THRUST PLATE
CRANKSHAFT THRUST SHIMS
CRANKSHAFT FRONT BEARING
CONNECTING ROD
CONNECTING ROD BEARING CAP
VALVE CLEARANCE .06 COLD
OIL LEVEL 6 QTS.
FLYWHEEL
CLUTCH PILOT BEARING
OIL PAN REAR BEARING OIL DRAIN
CRANKSHAFT REAR BEARING
OIL PAN GASKET
OIL INTAKE FLOAT AND SCREEN
OIL PAN DRAIN PLUG
OIL LEVEL GAUGE
CRANKSHAFT REAR CENTER BEARING
CAMSHAFT REAR CENTER BEARING
CRANKSHAFT FRONT CENTER BEARING
CAMSHAFT FRONT CENTER BEARING
VALVE LIFTER
VALVE ADJUSTING SCREW
VALVE SPRING SEAT
VALVE SPRING
VALVE GUIDE
VALVE

FIG. 313.—Section elevation, Studebaker engine

case. The supports for the crank-shaft bearings are a part of the cylinder block which contains the cylinder barrels. All of the castings are bolted together, with copper-asbestos gaskets placed between the joints. The water-cooling jacket is around the barrels as shown and extends into the cylinder heads. A water pump on the fan shaft is

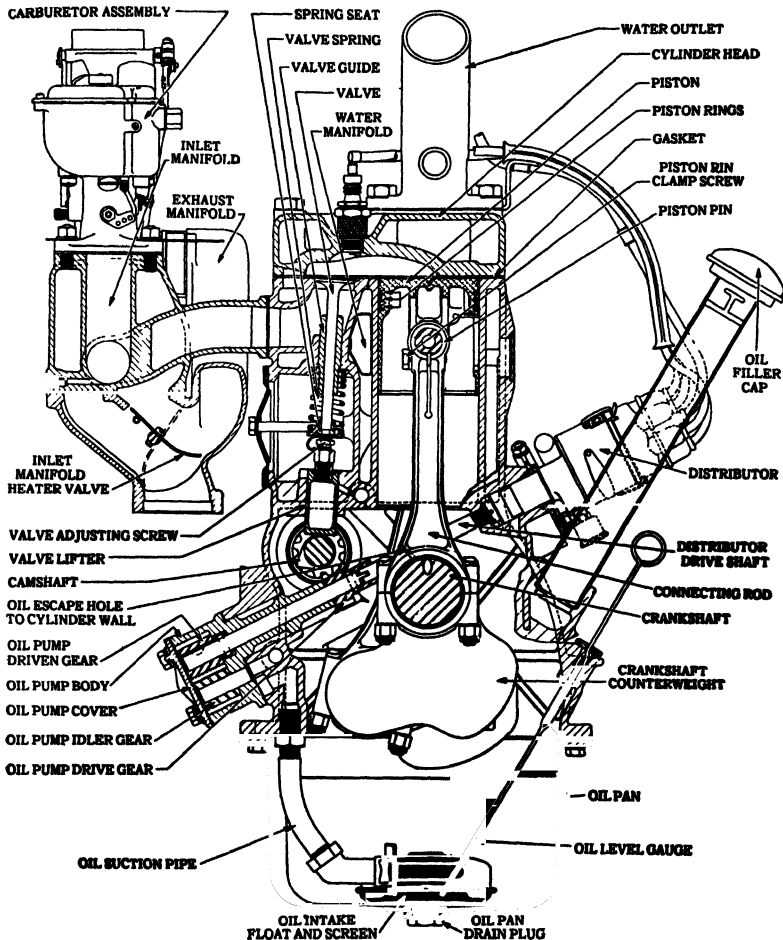



FIG. 314.—End section, Studebaker engine.

used to circulate the water through the jackets and radiator. The water flows from the front of the engine to the rear, where it passes up into the cylinder heads. It then returns to the front of the engine and passes out, to the top of the radiator.

The radiator used on this engine is of the fin and tube type. The water flows to the bottom as it is being cooled and thence to the pump.

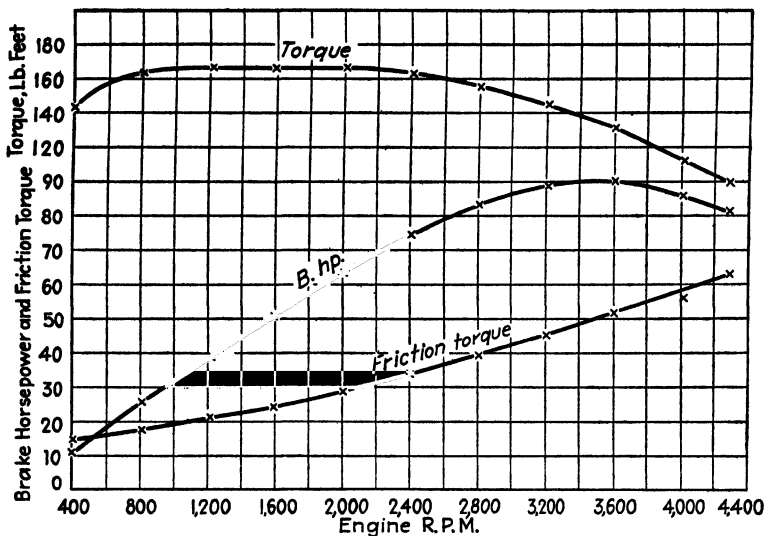



Fig. 315a.—Performance curves, Studebaker six-cylinder engine.

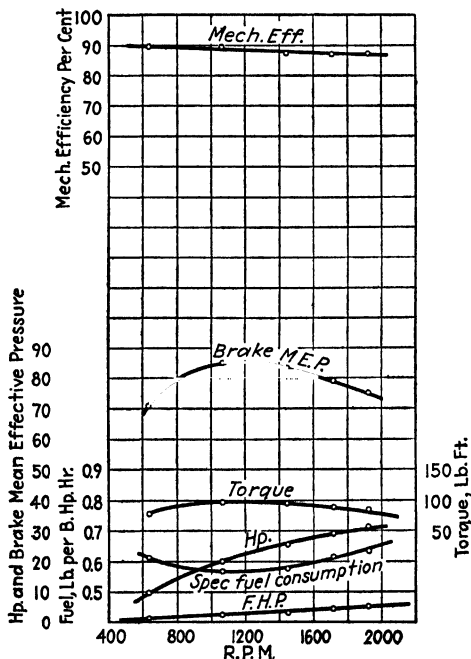



Fig. 315b.—Curves for results of Table 15-2, 1928 Chevrolet engine.

The air which cools the radiator is largely induced, at least at low engine speeds, by a four-bladed fan driven by a belt from the engine shaft. Water is automatically by-passed from the radiator until a predetermined temperature is reached. The automatic temperature control is by means of a thermostatic, sylphon bellows inside the radiator.

The cylinder diameter is 3¼ in. and the stroke 4⅜ in., giving a displacement of 217.8 cu. in. for the six cylinders. The compression ratio is 6.0 to 1 standard (cast iron), with 7.0 to 1 optional (aluminum). The performance curves for this engine are shown in Fig. 315*a*. It may be seen that the maximum brake horsepower is 90 at the speed of 3,600 r.p.m.

The combustion chamber of the Studebaker engine is of the L-head design. The intake manifold is placed above the ports leading to the intake valves, and two supply connections are provided to the cylinders. The exhaust manifold, with a connection to each cylinder is placed between the intake manifold and the cylinders. The flow of exhaust gas in the manifold is toward the middle of the engine. There the manifold curves down and then leads to the muffler at the rear.

The inlet valves each open at 15 deg. of the crank shaft before top dead center, and close at 49 deg. after bottom dead center, giving an intake period of 244 deg. The exhaust valves each open at 54 deg. before bottom dead center and close at 10 deg. after top dead center, which gives an exhaust period of 244 deg. The cam shaft turns at one-half of the crank-shaft speed.

The inlet valves are made of chrome-nickel steel, and the exhaust valves of sil-chrome No. 1 steel, a heat-resisting alloy. The pistons are made of an alloy, lyanite No. 132, and they are provided with a strut of Invar alloy, which gives lightness with strength. Each piston has three compression rings ⅛ in. wide. The crank shaft is made of drop-forged steel 0.38 to 0.48 carbon. There are four main bearings: No. 1—2½ by 1$\frac{5}{16}$ in.; No. 2—2½ by 1⅛ in.; No. 3—2½ by 1⅛ in.; and No. 4—2¼ by 1$\frac{27}{32}$ in. The crank shaft is balanced. Smooth running is obtained by the use of a torsional-vibration dampener.

A gear-type oil pump circulates lubricating oil under pressure to all bearings. The lubrication system is provided with an oil cleaner.

A Stromberg carburetor of the down-draft double-outlet type is used in the fuel system. A fuel pump (Fig. 316) is used to supply fuel to the carburetor. This pump, as shown, is operated by a lever arm which engages with a rotating cam. The lever arm, by means of a

link, moves downward the diaphragm push rod, compressing the operating spring. This action creates a suction over the diaphragm and draws through the suction check valve a supply of gasoline. As the link moves upward, the spring forces gasoline through the discharge check valve to the supply line of the carburetor. The auxiliary spring keeps the lever arm in contact with the cam.

The distributor of the ignition system and the oil pump are mounted on an inclined shaft driven by the cam shaft. Ignition spark occurs 2 deg. before top dead center. Other features include: water manifold to valves, forced lubrication to valve push rods, automatic choke by temperature control, automatic heat control and vacuum spark con-

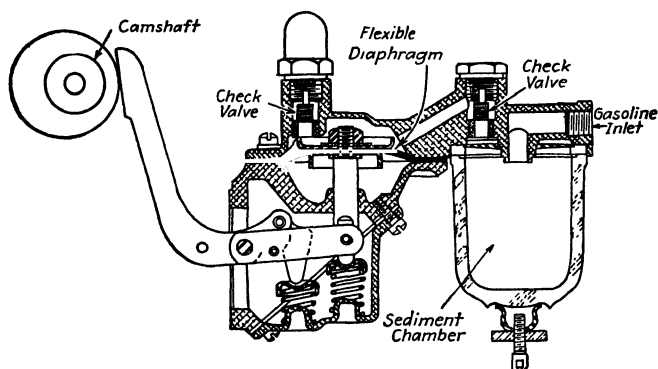



FIG. 316.—Fuel pump used on Studebaker engine. (A.C. fuel pump.)

trol. Outside of the engine are the planar suspension, overdrive, and twin-lever steering gear.

321. Carburetors.—Devices used with internal-combustion engines which have the important function of vaporizing the fuel and mixing it with the air consumed are called carburetors. They are used principally in connection with gasoline engines, though there are many kerosene carburetors in service.

The *ideal carburetor* is one which will give the proper air-fuel mixture ratio (pounds of air per pound of fuel) for all conditions of speed and load on the engine of which it is a part. It is often desirable to completely vaporize the fuel, though a carburetor which converts it to a fine mist and thoroughly mixes it with the air will perform its function satisfactorily.

The simplest form of gasoline carburetor employs a float chamber, which maintains constant level of fuel, and a nozzle projecting into an air passage leading to the intake manifold of the engine. The tip of the nozzle is placed slightly higher than the liquid level in the

float chamber in order to avoid leakage of fuel when there is no air flow. When air is flowing through a carburetor the resulting vacuum in the manifold causes the fuel to rise to the top of the nozzle and discharge into the stream of air. A carburetor of this type is termed the *plain tube carburetor*, and such a device, in operation, would give a richer mixture as the engine speed or air flow is increased. Most of the commercial carburetors are designed to overcome this tendency, and to give constant *air-fuel* ratios over the entire range of engine speed.

322. Carburetor Types.—Carburetors can, for the most part, be placed in one or more of the following classes:

1. Auxiliary air-valve type.
2. Expanding type.
3. Metering-pin type.
4. Compensating-jet type.

The first type is constructed with an auxiliary air valve placed between the jet nozzle and the manifold. This valve is held closed by a light spring, and as the engine speed and vacuum in the manifold increase, the spring tension is overcome to a point where the valve opens automatically and admits more air. By this means the air-fuel ratios tend to remain more constant.

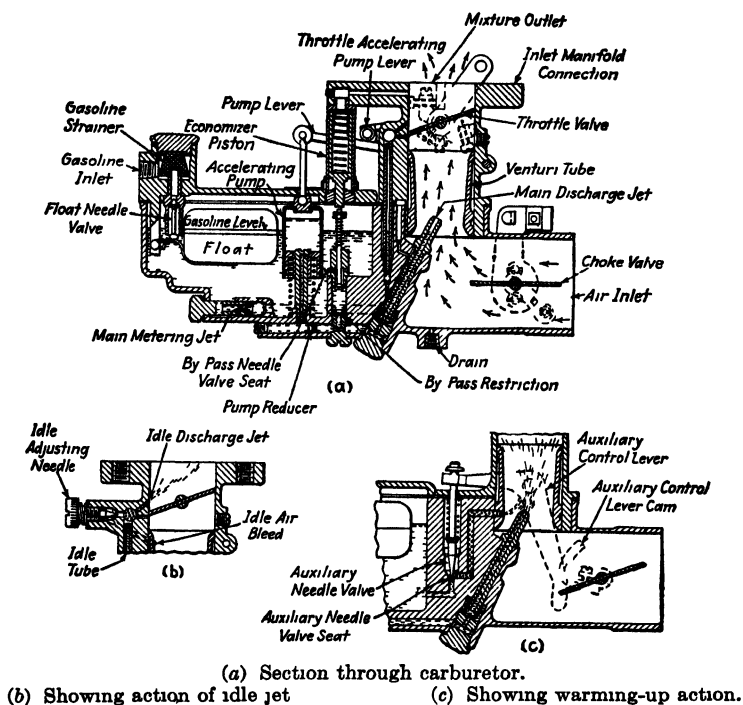
The *expanding type* carburetor employs a distinctive slide or rotary valve which automatically uncovers air jets with increasing air flow. This tends to give the proper proportions for air-fuel ratio with increased speed or load.

In the *metering-pin type* of carburetor, a tapered pin projects down into the nozzle. The size of the jet opening is regulated by moving the tapered pin up or down in the nozzle tube. This changes the amount of fuel flowing. Multiple jets with this principle are sometimes used.

The *compensating-jet type* of carburetor has two jets: a primary jet, which consists of a plain tube for delivering fuel into the path of the air, and the compensating jet, which is connected to the float chamber and also to an air-bleeder tube. At low speeds both nozzles function alike, delivering fuel into the air flow. With increased speeds, the compensating jet draws increased amounts of air through the bleeder tube, and this lessens the amount of fuel discharged through this jet. By this method the air-fuel ratio is kept constant.

323. The Stromberg Carburetor.—The Stromberg carburetor (Fig. 317) is an example of a plain-tube, gasoline carburetor provided with special features to insure a lean mixture at normal speeds,

but which supplies extra fuel for quick acceleration or heavy load conditions.

The gasoline level in the float chamber of this carburetor is kept constant by a float which controls an inlet needle valve. A strainer is provided in the inlet line. The fuel flows through the main metering jet to the main discharge jet and vertical tube that leads to the idle discharge jet. It is delivered into the path of the air through either or both of these jets, depending on the position of the throttle valve.

(*a*) Section through carburetor.
(*b*) Showing action of idle jet (*c*) Showing warming-up action.
FIG 317 —Stromberg U-type carburetor.

At speeds of less than 12 miles per hour the throttle valve is practically closed, and, in this case, all of the fuel enters through the idle discharge jet The adjustment of this jet is made as illustrated in detail in Fig. 317 *b*. With the throttle valve closed, a high vacuum exists in the manifold above, and fuel is thereby lifted up the idle tube and through the idle jet. Air is drawn from below the throttle valve, through the idle air-bleed tube to the needle valve, and thence into the idle discharge jet Turning the idle adjustment needle varies the amount of air drawn through the bleed tube for this purpose. When

this needle valve is opened more air is admitted, thus reducing the suction on the fuel in the idle tube. By proper adjustment of this needle valve, the best idling mixture can be obtained.

At speeds from 12 to 20 miles per hour, fuel is delivered into the path of the air flow through both the idle jet and the main jet. Above the latter speed practically all of the fuel is delivered through the main jet.

The main discharge jet has its outlet in the smallest section of the Venturi tube, which assures a high air velocity around the nozzle. This condition results in effective atomization of the fuel and mixing of fuel and air. Correct mixture is aided by bleeding air into the center tube of the main discharge jet.

Special features include the "economizer," which has the function of enriching the mixture under conditions of severe load, such as occurs when an automobile is ascending a steep hill. The economizer consists of a piston and a cylinder which has an air connection with the air outlet near the inlet-manifold connection. The vacuum in the manifold holds this piston in the position shown in Fig. 317 *a*. Under conditions of high load, the throttle valve is open and the manifold vacuum reduced considerably. The spring within the piston forces it down, which action opens the by-pass needle valve, and, in turn, increases the fuel flow to the main jet. When the load on the engine decreases, the vacuum in the cylinder again overcomes the spring force, and the by-pass needle valve is closed, bringing the fuel supply to the main jet back to normal.

In order to momentarily supply an extra-rich fuel-and-air mixture the throttle valve may be suddenly opened wide, which brings into action an accelerating pump. The throttle valve shaft is connected by a pump lever to the accelerating cylinder which contains an auxiliary supply of fuel. The quick opening of the throttle valve pushes down on the cylinder, thus putting a pressure on the fuel therein. This forces the piston at the bottom of the cylinder down and uncovers a port in the piston guide. The fuel then spurts through this port and out the main jet. As the pressure is relieved, the piston again covers the port, and the flow stops.

For starting a cold engine, an auxiliary jet is included, as shown in detail in Fig. 317 *c*. An auxiliary cam, attached to the shaft of the choke valve, opens an auxiliary needle valve when the choke is partially closed, as shown. This admits extra fuel through an auxiliary jet located in the side of the Venturi tube. Thus, a richer mixture for starting the engine is provided. The needle valve is automatically closed when the choke is again opened.

The Stromberg down-draft carburetor (Fig. 318) is illustrative of that type. The principal advantage is due to the effect of gravity, increasing the density of the fuel-and-air charge. Hence an increase

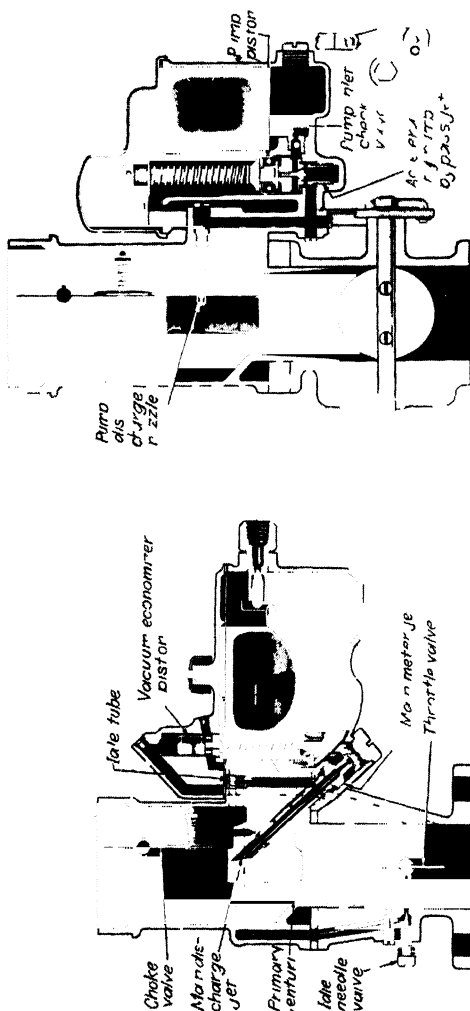



FIG. 318.—Stromberg down-draft carburetor

in engine power is effected. The down-draft carburetor has the same features as the up-flow type.

324. Methods of Supplying Fuel.—Methods for delivering fuel from the *storage tank* to the carburetor of an automobile engine employ gravity, vacuum or pressure. The Ford, model A car, with the gasoline tank in the cowl above the engine, is an example of the gravity-

feed system. The advantage of this is chiefly simplicity, and it avoids many of the troubles incident in the more complicated gasoline-feed systems. Only a short tube is required between the tank and the carburetor.

In the *vacuum system*, a small tank is placed above and near the carburetor. A vacuum in the tank, maintained through a connection from the intake manifold, is used to pull the fuel from the supply tank, which is usually at the rear of the car. From this small tank the fuel flows by gravity to the carburetor. The vacuum tank is constructed with suitable valves, etc., so as to maintain a ready supply of fuel available at all times. As it is emptied, a float operates the valves so as to refill it to a definite level. Thus, the operation of the tank is intermittent.

Pressure systems include two distinctly different methods of feed. The older system provides an air pressure in the storage tank and thus forces the fuel to the carburetor. The pressure is produced by an air pump geared to the engine. A small hand pump is provided for starting.

The present pressure system employs a mechanically or electrically operated fluid pump placed between the storage tank and the carburetor.

325. Ignition.—The gasoline engine for automobiles is equipped with an electrical system for ignition of the fuel, starting, lighting and the horn. Because of the latter functions, the system generally includes a storage battery, with a generator for charging it. Engines with "dual ignition" are provided with both battery and magneto ignition. This, however, does not mean "multiple ignition," which refers to ignition systems providing more than one spark plug in each cylinder of the engine.

326. Make-and-break Ignition System.—One of the early developments of electrical ignition was the make-and-break ignition, still in use although generally superseded by other systems. The arrangement in the make-and-break system is such that a movable contactor can make electrical contact inside the engine cylinder. The movable contactor makes or breaks contact by rotating through a small angle, and the movement is derived from a cam mechanism driven by the crank shaft. The stationary contactor is properly insulated from the cylinder, and the circuit includes a source of current and low-tension choke coil in series. The spark for ignition occurs when the circuit is broken. At the proper time for ignition, the movable contactor is released and snaps apart from the stationary contactor, thus forming the gap across which the spark jumps.

327. Jump-spark Ignition System.—A system called the *jump-spark ignition,* often used for automotive engines, is illustrated by the diagram in Fig. 319. A typical high-voltage jump-spark system has two circuits; that is, for the primary or low voltage and the secondary or high voltage. The low-voltage circuit includes the source of current, the timer which opens and closes the circuit at the proper time, and the primary windings around the iron core of an induction coil. The high-voltage circuit derives its current from the secondary windings around the same core. When current flows in the primary windings of an induction coil a magnetic field is set up, which converts the iron core of the coil into an electromagnet. The magnetic field cutting the conductors of the secondary circuit slowly induces a high tension in this circuit. Consequently, the spark occurs when contact is first made. At the instant that the timer breaks the primary

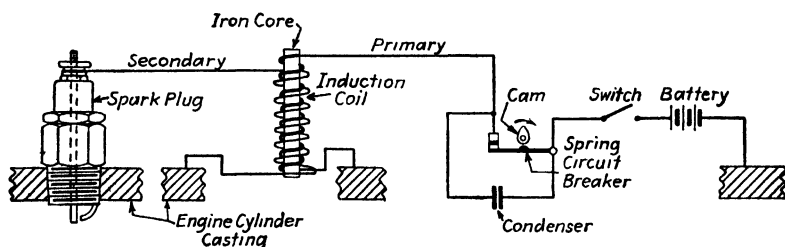



FIG. 319.—Diagram of high-tension ignition system, using battery current.

circuit, the collapse of the magnetic field suddenly induces a much higher voltage in the secondary circuit and a spark jumps across the gap between the spark-plug points. During the collapse of the magnetic field a current is also induced in the primary coil, and a condenser is placed across the breaker points to absorb this current and prevent the points from sparking. The function of the condenser is to absorb or cushion the surge of current that occurs when the circuit is broken, which is similar to the action of the air chamber of a hydraulic pump. For a multiple-cylinder engine, a timer having a cam for each cylinder would be required in the primary circuit, and an automatic switch (distributor) in the secondary circuit.

With the type of ignition system just described, the intensity of the spark decreases at high engine speeds, because of the shorter time interval allotted for building up the magnetic field. Another factor which tends to reduce the spark is high cylinder compression which creates a denser medium to be punctured by the spark. The use of more powerful coils and devices for increasing the time for building up the field are the lines along which this type of ignition system is being improved.

Figure 320 shows a diagram of a high-tension ignition circuit using a low-tension magneto as the source of current. The magneto contains an armature rotating in the field of a permanent magnet, and the primary circuit of the induction coil obtains low-voltage current from the armature winding. The secondary circuit, as in Fig. 319, includes the secondary winding of the induction coil and the spark plug. The vibrator or breaker arm is made from a strip of spring steel and tends to hold the two contacts together, as shown by the figure. When the rotating arm of the timer closes the primary circuit a voltage is in turn built up in the secondary circuit. The magnetic field produced by this current at once attracts the spring

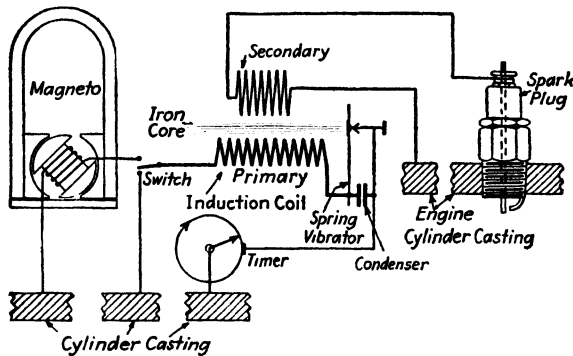



FIG. 320.—Diagram of high-tension ignition system, using magneto current.

vibrator and the primary circuit is opened. Here, again, the sudden collapse of the field induces a high voltage in the secondary circuit, and a spark jumps the gap of the spark plug. The condenser is connected across the breaker points to reduce the arcing tendency at this point. When the primary circuit is broken the vibrator returns to its former position and closes the circuit again. The sparking in the cylinder recurs at short intervals as vibrations, until the timer, in its rotation, opens the primary circuit.

In what is called the *high-tension magneto*, the primary and secondary windings of the induction coil are wound around the magneto armature, and the separate spark coil is not necessary. For a multiple-cylinder engine a means (distributor) for distributing the current to the various cylinders at the proper time is used in the high-tension circuit. Magnetos of this type are sometimes used on tractor and truck engines, and for airplane ignition.

The most common ignition system used on present-day automobile engines employs a 6-volt storage battery and a generator. The generator is driven by the engine, and the battery "floats" in the circuit.

The current for the lights, horn, starting motor and ignition is furnished by the battery, which, when the speed of the car exceeds approximately 10 miles per hour, is charged by the generator. A cut-out relay is provided to prevent the battery from discharging through the generator when running slow or when the engine is stopped. The cut-out relay is illustrated by the diagram in Fig. 321. It may be noted that when the generator is running, the core of the differential coil is magnetized. This holds the breaker contacts together, which completes the circuit through the battery and allows charging current to flow. At slow speeds, and when the generator stops, the generator voltage drops below that of the battery, and the magnetism of the

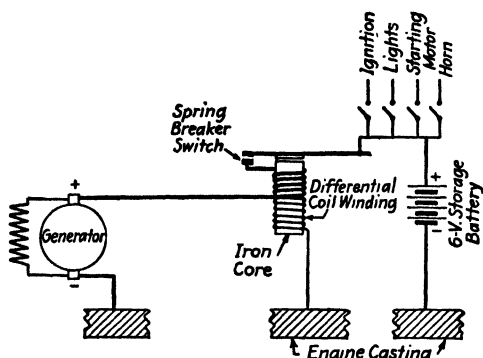



FIG. 321.—Diagram of cut-out relay circuit.

core is neutralized as a result of the differential winding. The contacts are then separated, thus isolating the generator from the battery.

Regulation of the charging current, when the generator is running at high speeds, must be provided in order to protect the battery from excessive voltage. Common methods of obtaining such regulation include the use of a *vibrating-type regulator* or *third-brush regulation.* Figure 322 shows a diagram of the generator-battery circuit only, in which the vibrating regulator is used. On starting, the first revolutions of the generator armature produce a slight voltage in the armature windings, a result of residual magnetism in the field cores. Current then flows in the generator field windings, the circuit being complete through the contacts *A*, which are held closed by a spring. This, in turn, strengthens the field magnetism, and the generator voltage is built up, which, when reaching a value of approximately 7 volts, closes the contacts *B* of the cut-out relay. Charging current then flows through the battery. At high speeds the current produced by the generator increases to a point where the magnetism of the core of the regulator is sufficient to open the contacts *A*. This causes the

field current to flow through the resistance unit, thus weakening the generator field magnetism and causing a corresponding drop in the generator voltage. As a result the magnetism in the core of the regulator is weakened, and the spring draws the contacts *A* together again, repeating the cycle. Thus, a series of vibrations is set up, and the current flow is thereby controlled.

If the regulator is placed in parallel with the generator, instead of in series as shown, regulation is accomplished by changes in the voltage; not by changes in the charging current.

In third-brush regulation the third brush, which connects to one end only of the shunt-field winding, is placed on the commutator intermediate between the two main brushes. When the generator is running at low speed, the magnetic field produced is approximately straight from one pole piece to the other. The voltage generated

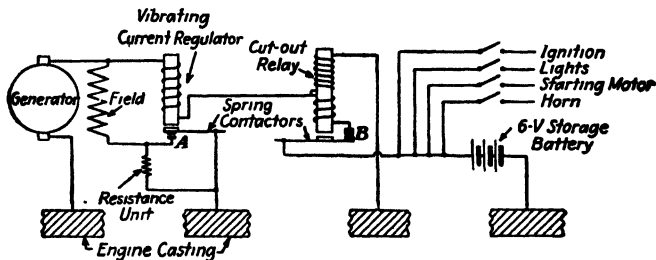



FIG. 322.—Diagram of vibrating type of current-regulator circuit.

in the shunt field depends upon the lines of force intercepted by the third brush.

As the generator speed and charging rate increase, the magnetic field is distorted so that the field intercepted by the third brush becomes weaker, and consequently a lower voltage is generated. The result is a dropping off of the voltage generated as the speed increases. Consequently, there is an automatic regulation of the current output.

The position of the third brush is easily adjustable thereby permitting the regulation of the normal charging rate.

328. Automotive Starting Devices.—Automotive *starting devices* usually include a series-wound motor, pinion and gear and a suitable clutching mechanism. Chain drives, in place of gears, are also used. The act of the motor is to turn the flywheel of the engine through several cycles until complete operation begins. Starting is a severe drain on the storage battery, especially at low temperatures. For this reason it is often necessary to use lighter lubricating oil in cold weather.

329. Stationary Gas Engines.—Stationary gas engines have found increasing use with the increased availability of natural gas. Today pipe lines spread through 38 states, a 26-in. line running into Chicago and a 22-in. line to Detroit. Refineries use stilled gas in gas engines rather than under boilers. Sewage-disposal plants use sludge gas in gas engines to produce power at practically no fuel cost. Other uses are found for stationary gas engines in industries, owing mainly to the cheaper supplies of natural gas.

Normally the large gas engines are double acting, horizontal and, except in small sizes, of the tandem cylinder. This type of engine has one crank and connecting rod, one main frame, and one main and one outboard bearing, yet is equivalent to four single-acting cylinders except in smoothness. The twin-tandem is likewise equivalent to eight single-acting cylinders. The horizontal tandem seldom runs over 130 r.p.m. on compressor drives and 150 to 160 r.p.m. on generator drives.

The world's largest gas engine is a horizontal, double-acting, twin-tandem, four-cycle, 6,600-kw., engine compressor at the South Chicago Works of the Illinois Steel Company.

The largest vertical engines are units of 1,800-hp. capacity at 225 r.p.m. These 12-cylinder engines attain an economy of 9,700 B.t.u. per brake horsepower per hour. High-speed (900 to 1,200 r.p.m.) vertical gas engines are a composite of automotive and Diesel designs, normally four cycle, single acting.

Fuel economy runs about 12 cu. ft. of 1,000 B.t.u. of gas for small engines, dropping to about 10 cu. ft. for large engines.

330. Oil Engines.—The modern oil engine can be made from about 3 hp. up to 22,500 hp. Most large engines are two-stroke cycle and most small ones four-stroke cycle. Piston speeds vary from about 800 to 1,600 ft. per minute in small engines to 1,000 to 1,200 ft. per minute in large engines.

In new designs air injection has been almost eliminated in large engines, except for installations at a plant containing engines of this type or where a very poor fuel is to be burned. All large oil engines are vertical engines. There are no four-cycle double-acting engines, but a considerable number of large two-cycle units are double acting.

In large oil engines, fuel is sprayed by direct injection into the combustion space. Generally, fuel nozzles have many small orifices approximately 0.015 to 0.035 in. in diameter. The combustion space is, in most cases, designed to create turbulence to assure good mixture of fuel and air. To a large extent the smaller high-speed oil engines are patterned after automotive design.

TABLE 15-3.—TYPICAL GAS-ENGINE PLANTS

Owner	Location	Service	Equipment driven	Builder	Number of engine units	Hp each unit	R p m	Number of cylinders	Cylinder bore, in.	Stroke, in.	Horizontal or vertical	Single or tandem	Cycle	B.t.u. per b.hp-hr	Type of fuel	B.t.u. per cu. ft. fuel
Panhandle East Pipe Line Co.	Houstonia, Mo.	Gas	Comp	C B.	4	1,300	125	4	24	36	H	TT	4	10,000	Nat	1,000
	Pleasant Hill, Ill.	Gas	Comp	Wor	2	1,300	125	4	24	36	H	TT	4	10,000	Nat	1,000
	Liberal, Kan.	Pwr	Gen	C. P	2	188	400	4	11¼	13¼	V	S	4	10,000	Nat	1,000
Corning Glass Co	Corning, N Y	Pwr	Gen	I R.	1	1,200	225	6	22	26	V	S	4	10,000	Nat	1,000
Sun Oil Co	Marcus Hook, Pa.		Pump	Ster	1	500	1,200	8	8	9	V	S	4	10,000	Ref	1,600
Peoria	Illinois	Mun Sewage	Blow	Wor	1	535	327	6	15	17½	V	S	4	9,500	Slu	550
			Gen	Wor	1	300	400	5	12¼	14½	V	S	4	9,500	Slu	550
Birch Oil Co	Brea, Calif.	Ref	Alt	Cla	1	300	500	6	9	11	V	S	2		Nat	1,000
					1	250	500	4	9	11	V		2		Nat	1,000
Wilson-Snyder Mfg. Co	Braddock, Pa.	Ref	Pump	C. B	1	235	200	2	17	20	H	Tw	2	10,000	Ref	1,500
				C B	3	155	250	2	13½	18	H	Tw	2	10,000	Ref	1,500
Bi-Rite Market.	Los Angeles, Calif.	Store Pwr	Gen	Wau	1	67	1,000	6	5	5½	V	S	4	11,200	Nat	1,120
					1	36	1,000		4½	5¼	V	S	4	11,200	Nat	1,120
Cedar Rapids	Iowa	Sew Disp	Gen	Wor	1	210	300	5	12⅛	14½	V	S	4	10,000	Slu	600

NOTE.—Abbreviations: see also Table 15-5, p. 528; C P = Chicago Pneumatic Co; Cla = Clark Bros., Ster = Sterling Engineering Co; Wau = Waukesha Motor Co.
Alt = alternator; Blow = blower; Comp = compressor, Gen = generator; Nat = natural gas, Ref = refinery gas; Slu = sludge or sewage gas; Tw = twin; S = single; TT = twin tandem; Mun = Municipal.

331. Oil-engine Types.—The Diesel engine, as previously stated, originated in Europe, and during the early development it was built in this country under the European licenses.

The main developments have aimed at reducing the weight per horsepower output and obtaining as effective atomization and com-

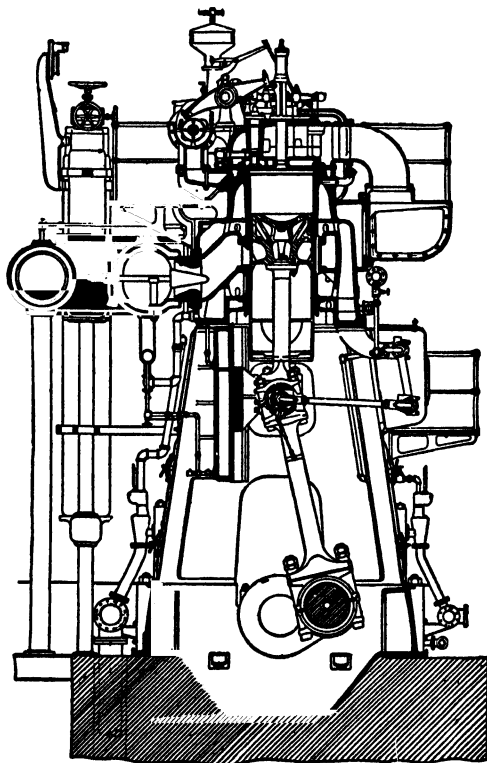



FIG. 323.—Sectional view of a Nordberg Diesel engine, 3,750 hp. capacity.

bustion with solid injection as have resulted with air injection. The following list gives various features of design that have been used in improving the Diesel engine:

1. Two-stroke cycle.
2. Double-acting engine.
3. High-speed.
4. Improved spray-nozzle design.
5. Improved combustion-chamber design.
6. Lighter and stronger materials.

Figure 323 shows a sectional view of a typical large Diesel engine. This is a six-cylinder, two-stroke cycle, single-acting engine with 29

by 44-in. cylinders. The operating speed is 125 r.p.m., and it is rated at 3,750 b.hp. Air-fuel injection is used, which employs a three-stage air compressor, and a scavenging pump, driven from the crank shaft, supplies air to expel burned gases from the cylinders and replace these gases by air for the combustion during the succeeding cycle. The scavenging air valves, fuel-injection valve and the air-starting valve are all in the cylinder head. The piston performs

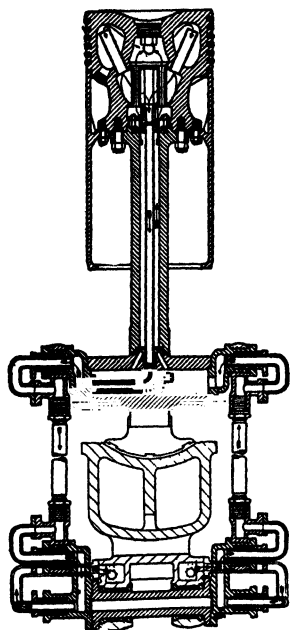



Fig. 324.—Showing piston cooling-system design, Nordberg Diesel engine.

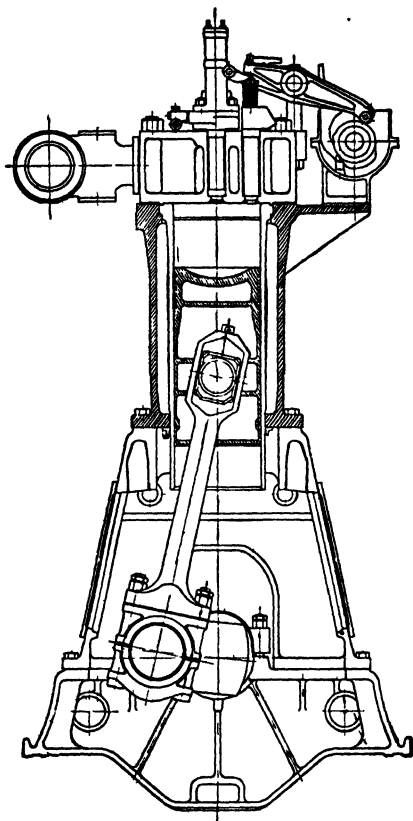



Fig. 325.—Busch-Sulzer four-cycle Diesel engine.

the function of the exhaust valve, covering and uncovering the exhaust ports in the cylinder barrel. Crosshead and connecting-rod construction is used for connecting the piston to the crank shaft. Both the cylinder and the piston are water cooled, the water having access to the piston through a swinging-joint piping arrangement. This connection is to the hollow wrist pin of the crosshead, and the path of the water is as shown in Fig. 324. It flows through the hollow piston rod to the piston, and, after circulating through the internal cavities, returns to the crosshead through a tube in the piston rod. The outlet from

the crosshead is at the opposite side from that at which the water enters.

Another style of Diesel engine is illustrated in Fig. 325, which shows a sectional view of an engine manufactured by the Busch-Sulzer Bros. Diesel Engine Co. The striking differences between this and the Nordberg engine, just described, is the trunk-piston construction used and the four-stroke cycle of operation. Both the intake and the exhaust valves are located in the head and are operated by rocker arms from cams on the cam shaft. The fuel-injection valve and the air-starting valve are also located in the cylinder head. This

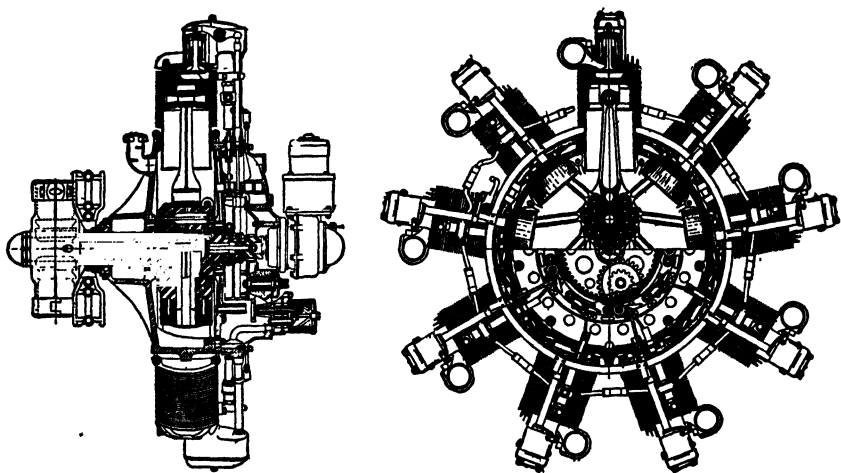



Fig. 326.—Packard radial Diesel, airplane oil engine.

engine has the integral three-stage air compressor at one end of the shaft.

A significant development in this field is the application of the oil engine to aviation. Figure 326 illustrates the Packard, nine-cylinder aviation oil engine which operates on the four-stroke cycle and has a power capacity of 200 hp. The normal speed is 1,950 r.p.m. and the weight is 600 lb. The cylinders are $4\frac{13}{16}$ by 6 in. Each cylinder has one large valve which performs the functions of both intake and exhaust. This valve is located in the head and is operated by a push rod and rocker, from a cam ring driven at one-eighth of the engine speed. This valve, during operation, remains open for both the exhaust and intake periods. The burned gases discharge through a port in the cylinder head to the atmosphere, during the exhaust stroke of the piston. This same port provides a passage for the flow of air to the cylinder during the suction stroke. Twelve springs are

used for each valve to prevent the possibility of trouble by breakage and by any surging effect in the valve spring. Solid-injection system of fuel supply is used on this engine, with an individual plunger for each cylinder. The fuel pump is combined with the spray valve, which is located at the top of the cylinder, and the charge of fuel is sprayed into a recess in the piston. Figure 327 shows a diagram of the fuel-pump mechanism by which the speed of the engine is controlled. The plunger is actuated by the cam ring, through a rocker arm and push rod. The amount of fuel injected into the cylinder depends upon the stroke of the plunger, which is regulated by swinging the push rod to various positions on the rocker arm. It may be seen that the farther the push rod is swung away from the pivot of the rocker arm the longer will be the effective stroke of the fuel-pump plunger. The time of injection can be altered by changing the position of the cam ring, relative to the engine shaft. Each individual pump cylinder is supplied with fuel from an auxiliary pump. This fuel is supplied under pressure, for safety in preventing air from entering the suction lines and binding the pump.

FIG. 327.—Diagram of fuel-pump mechanism, Packard Diesel engine.

Methods used for delivering fuel into the cylinders of oil engines are called (1) *air injection*, and (2) *solid* or *mechanical injection*. In the former, the fuel is injected by means of high-pressure air, while in the latter the fuel, alone, is forced to the cylinder through a suitable nozzle.

332. Discussion of the Air-injection System.—The original commercial Diesel engine and many engines of today employ the air-injection method, which is effective in obtaining good atomization and complete combustion of the fuel oil in the cylinder. There is, however, a disadvantage in this design as it necessitates an air compressor, which requires power and additional maintenance.

Air compressors may be driven from an independent motor or engine, or they may be built integral with the engine. In many installations, particularly for marine engines, both independent and integral compressors are used to prevent compressor trouble from interfering with the operation of the engine. On small, single-cylinder engines a two-stage compressor is commonly used, but on sizes of over 100 hp., the use of a three-stage compressor is the usual practice.

A three-stage integral compressor is illustrated in Fig. 328. It has a rather long piston of separate diameters which operates in three corresponding cylinders. The first stage or low-pressure cylinder holds the piston of largest diameter, the intermediate stage is the annular volume around this piston, and the high-pressure stage is at the top, above the piston of smallest diameter. The air is com-

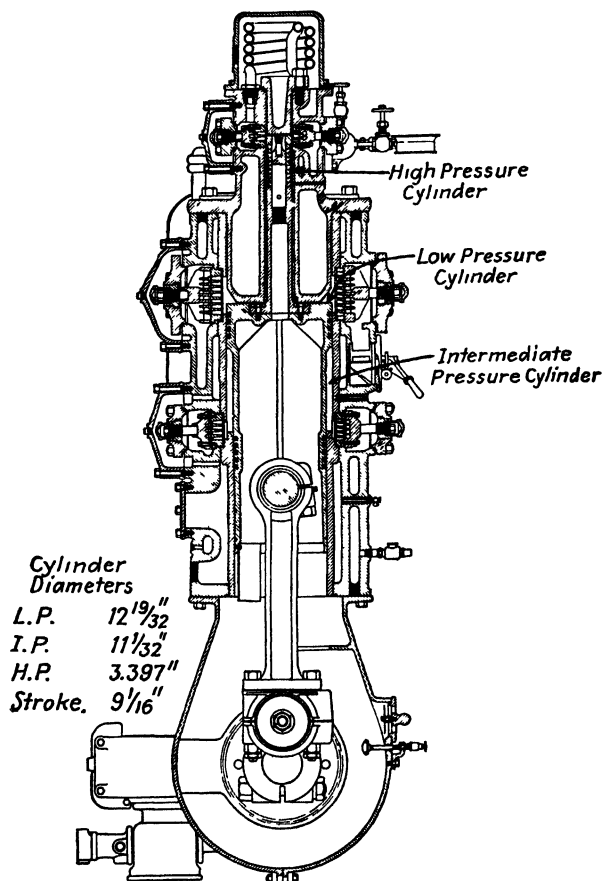



FIG. 328.—Three-stage integral air compressor.

pressed in the low- and high-pressure stages during the upward stroke of the piston and in the intermediate stage during the downward stroke. The cylinders are water cooled, and intercooler coils are provided to cool the air discharged from each compression stage. This reduces the volume of the air and improves temperature conditions for the operation of the valves. Valves for this purpose are commonly of the ring or feather check type, which are similar in their operation.

Average values of the pressures leaving each cylinder are as follows:

1. Low pressure........................ 60 lb. per square inch.
2. Intermediate pressure............... 230 lb. per square inch.
3. High pressure....................... 1,000 lb. per square inch.

A compressor for a 1,000 hp. Diesel engine has a maximum capacity of 450 cu. ft. per minute, and a maximum attainable pressure of 1,300 lb. per square inch. This is about double the capacity required, and in normal operation the inlet of the low-pressure cylinder is restricted. Special alloy steel, air-storage bottles capable of withstanding a pressure of 1,000 lb. per square inch are a part of the equipment and hold the air supply used for starting the engine. For purposes of safety, all high-pressure air tubing is made of copper.

In the air-injection system the fuel is injected in nozzles of two types, (1) the closed nozzle and (2) the open nozzle. Both attain the result of more or less perfect fuel atomization and discharge of the fuel, in a fine spray, into the cylinder against high-compression pressure. Dr. Diesel originally adopted mechanical injection, but he discarded this when experiments showed that more efficient atomization resulted from the aid of high-pressure air. He then developed the closed-nozzle valve which, with refinements, is widely used in many modern Diesel engines. The typical closed valve is mounted over a short nozzle in the center of the cylinder head. Immediately over the nozzle is the atomizer, the purpose of which is to break up the liquid fuel into a fine spray before it enters the cylinder. The atomizer consists of a number of rings or washers containing many minute perforations. Below these rings are spiral grooves, or other tortuous passages leading to the nozzle. There is an annular space around the valve stem above the atomizer, which is connected to the compressor discharge. It is thereby filled with air at approximately 1,000 lb. per square inch pressure. In the operation, the fuel pump forces a metered amount of oil into the space directly above the atomizer. When the valve is opened by the cam mechanism, the fuel is broken up in passing through the atomizer, and, entrained by the injection air, enters the cylinder when the piston is at dead center, which is the point of maximum compression. The expansion of the injection air entering the cylinder further increases the fineness of atomization and improves combustion conditions. This expansion, however, results in cooling and has a deterring effect, but this effect is only momentary as combustion starts immediately, bringing the cylinder temperature to above 2000°F. The ignition of the fuel is effected by the high temperature in the cylinder resulting from com-

pression. Compression of the air in the engine cylinder to 500 lb. per square inch, aided by the heat from preceding cycles, gives a temperature of from 1000 to 1400°F.

The open-type, air-injection nozzle is often used with horizontal engines. In the operation of this nozzle, a measured charge of oil is pumped into a small recess in the fuel valve when the piston is at the crank-end dead center previous to compression. Consequently, the fuel pump does not work against a high cylinder pressure and for this reason can be of lighter construction. This permits more sensitive control by the governing mechanism. The recess which receives the charge of fuel is open and between the air valve and cylinder. At the end of the compression stroke of the engine piston, the air valve is automatically opened and the injection air discharges through the recess, picks up the fuel, and forces it into the cylinder. The advantage of this type of nozzle is that the main valve passes air only and, therefore, does not tend to "gum up" and leak. On the other hand, it does not give as complete atomization as is attained with the use of the closed type of nozzle.

333. The Waukesha Oil Engines.—The Waukesha Motor Company manufactures two oil engines which have various uses as the drive for such equipment as air compressors, electric power plants, excavators, tractors, locomotives, marine engines, pumps, motor vehicles and many others. These two types of engines are named the *Waukesha Hesselman Oil Engine,* and the *Waukesha Comet Diesel.* The operation and principles of these engines will be discussed.

The *Waukesha Hesselman* engine (Fig. 329) is a low-compression, electric-spark-plug-ignition oil engine using solid injection of fuel oils, commonly termed "Diesel fuels." The compression ratios used vary from 5.75 to 1, to 6.25 to 1. The name of the engine is from its inventor K. J. E. Hesselman of Stockholm, Sweden. The intake valve is located in the cylinder head. Fresh air is drawn into the cylinder during the suction stroke, and receives a rotary motion about the vertical axis of the cylinder by a ramp-type inlet port in the cylinder head. The resulting air velocity is high enough to insure a continuance of this whirling motion during the compression stroke. The fuel charge is injected about 50 deg. before top dead center. The turbulent air whirl during the remaining part of the compression stroke serves to mix thoroughly the finely atomized fuel with the air, and to carry this combustible mixture past the spark plug where it is ignited at approximately 15 deg. before top dead center. The spark plug and fuel-injection valve are on opposite sides of the cylinder. After the start of combustion the rest of the cycle is like the Otto-engine cycle.

Neither the air nor fuel is preheated outside of the combustion chamber. Hence the volumetric efficiency and power developed is high compared with engines with preheating as a part of the cycle.

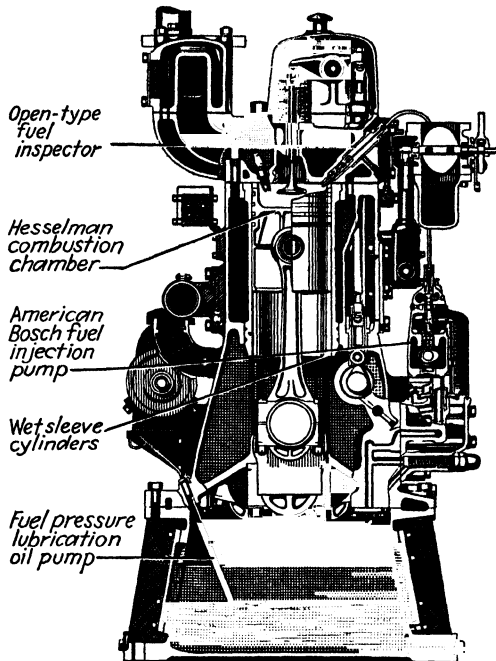



FIG. 329.—Waukesha Hesselman oil engine, end section.

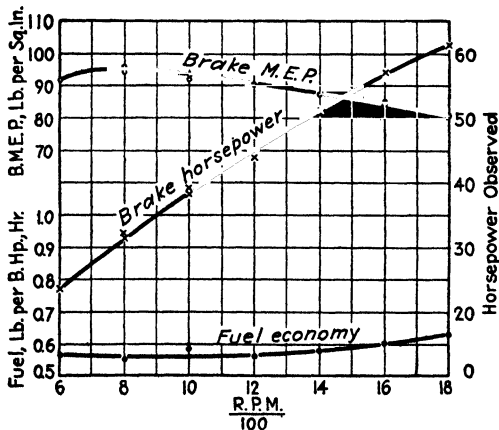



FIG. 330.—Waukesha-Hesselman engine test.

A brake mean effective pressure of 104 lb. per square inch has been obtained on an engine of this type, with 6½ in. diameter of cylinder. Figure 330 shows actual test results.

The *Bosch fuel pump*, with one plunger for each engine cylinder, is used in connection with a special solid-injection nozzle. This pump (sectional view of one pump cylinder shown in Fig. 331) has no suction valve, and a spiral groove on the plunger acts as a pressure relief valve toward the end of the discharge stroke, by connecting the space above the piston with the suction chamber. When the helical edge of the plunger uncovers the inlet port on the right, the fuel remaining above the plunger escapes back to the suction chamber. The governor moves a control rod on which teeth are cut to link with gears on each cylinder sleeve. When the control rod is moved by the throttle linkage, all of the plungers are turned in unison. The turning of the plunger changes the time of pressure release and the quantity of fuel delivered per stroke. A cam pushes the plunger upward on the discharge stroke, and a spring returns the plunger on the suction stroke. The discharge passes through a spring-loaded check valve at the top. The pump is a constant-stroke, variable-discharge pump.

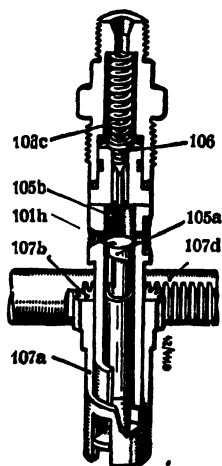



FIG. 331.—Bosch fue pump.

Figure 332 shows by diagram the method of controlling the air-fuel ratio at different loads and speeds. A butterfly valve throttles the

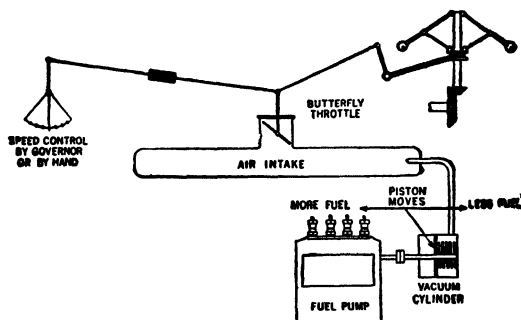



FIG. 332.—Air-fuel control, Waukesha engine.

air entering the intake manifold. When the butterfly valve closes, either by hand throttle or by automatic governor, a vacuum is produced in the manifold proportional to the restriction to the air flow. The variation in the intake-manifold vacuum is used to actuate a piston connected to the fuel-pump control rod. This control rod contains the racks which govern the volume of fuel discharged by the individual plungers. Hence a movement of the butterfly valve to

decrease or increase the air flow will have the same effect on the volume of fuel supplied, and a correct air-fuel ratio is maintained.

The injection nozzle contains three check valves in series to prevent dribbling and prevent combustion-pressure kick-back into the injection

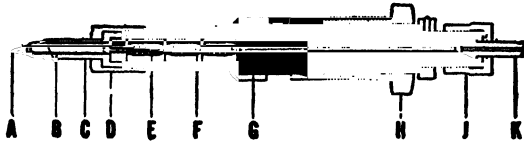



FIG. 333.—Injection nozzle, Waukesha engine.

line. Referring to Fig. 333, starting at K, fuel enters the nozzle, passes through the check valves, and then along the two V-feed grooves

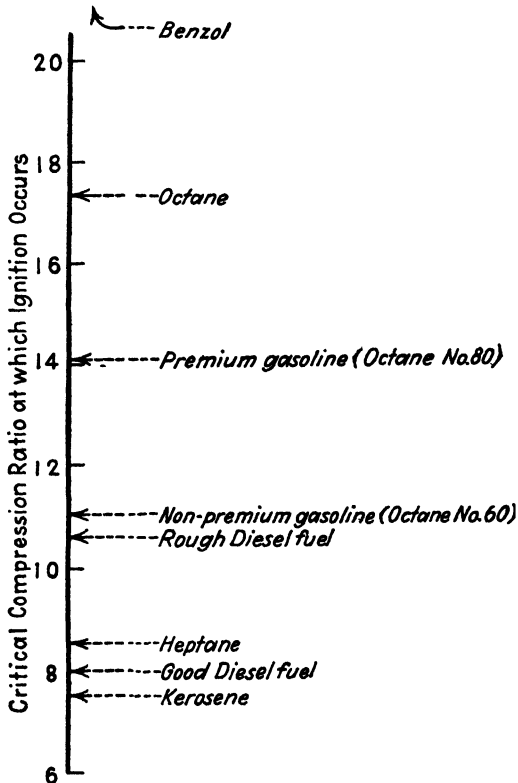



FIG. 334.—Range of ignition quality covered by liquid fuels of various types.

in a special plug within the outer casing. At the end of these grooves is an atomizing recess which gives the fuel a swirling motion before it passes through the spray holes. There are two spray holes, about 30 deg. apart, one directed against the air swirl in the cylinder, and

one with it. The injection pressure required varies from 650 to 1,000 lb. per square inch. The spray-hole sizes vary from 0.015 in. for small engines to 0.030 in. in larger ones. The nozzle tip is made of "nitralloy" steel.

The engine is started by spraying, from a primer, gasoline into the intake manifold. The overhead valve construction causes the fuel to flow into the cylinder when the engine is cranked. The cupped piston head holds gasoline not vaporized or atomized. The compression temperatures are high enough to vaporize sufficient fuel to form a combustible mixture at the spark plug. After starting, the engine continues on fuel oil.

Fuels should have high enough viscosity to provide pump-plunger lubrication but must atomize properly and must be free from asphalt, tar, gum, acid, or sulphur, which might cause fouling of spark plugs, clogging of nozzles, and gumming of valve stems. Probably the most satisfactory fuels are the refined No. 2 or No. 3 furnace oils, with viscosity range of 35 to 50 Saybolt. However, a very wide range of fuels, including kerosenes, light distillates, and even gasoline, can be used because of the low compression ratio (Fig. 334), and because of the possibility of adjusting the time of the fuel injection. Because of the nature of combustion, detonation is not likely to be serious. Fuel is in the cylinder for only a few degrees of crank angle before ignition, and is mostly in a liquid state in the space ahead of the flame travel. Vaporization tends to restrict or eliminate detonation. Combustion is practically completed at 18 deg. of crank angle after top dead center. The peak pressure in the cylinder at the completion of combustion is approximately 650 lb. per square inch.

The Waukesha-Hesselman engine is made in five sizes in the six-cylinder model and six sizes in the four-cylinder model. The largest and smallest engine in each model is included in the following table giving general specifications:

TABLE 15-4.—WAUKESHA-HESSELMAN AND WAUKESHA-COMET OIL ENGINES

Model	Rated horsepower	Speed at rated horsepower, r.p.m.	Bore and stroke, in.	Total displacement, cu. in.	Approximate engine weight, lb.
6-cylinder....	280 to 300	850 to 950	$8\frac{1}{2} \times 8\frac{1}{2}$	2,894	7,950
6-cylinder....	60 to 75	1,800 to 2,500	$3\frac{3}{4} \times 4\frac{1}{4}$	282	750
4-cylinder....	100 to 110	950 to 1,050	$6\frac{1}{2} \times 8$	1,060	3,200
4-cylinder....	30 to 40	1,400 to 1,800	$3\frac{3}{4} \times 4\frac{3}{4}$	210	715
		Waukesha-Comet			
6-cylinder....	140	2,200	$5 \times 5\frac{1}{2}$	648	2,165
6-cylinder....	100	2,200	$4\frac{3}{8} \times 5\frac{1}{8}$	462	1,675

The *Waukesha-Comet* oil engine, a compression-ignition engine is made in three sizes, the largest and smallest being shown above. Figure 335 shows an end section of this engine. The Ricardo spherical combustion chamber is one of the features of this design. This chamber block is above and to the left of the cylinder and is a separate insert surrounded by an air space, so that it retains the heat from each combustion, to return it to the succeeding charge of air. The large mass of metal also aids this heat storage.

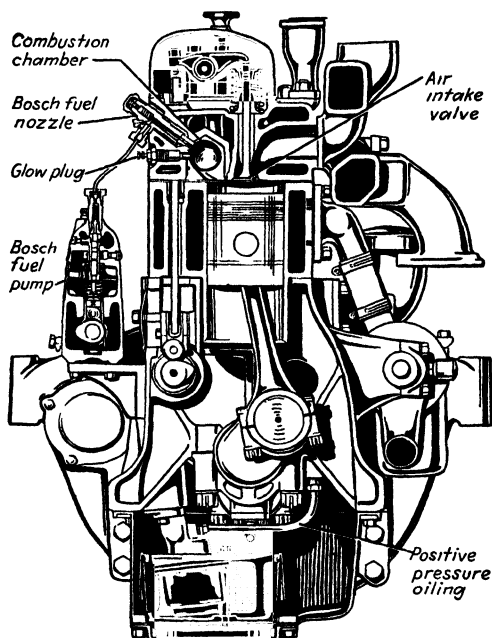



Fig. 335.—Waukesha comet oil engine.

At top dead center there is small linear clearance between the piston and the cylinder head, and about 80 per cent of the intake air is forced into the combustion chamber through the tangential opening. This creates in the chamber a turbulent whirl which sweeps from the nozzle tip each atom of fuel as rapidly as it is injected. Thus a good air-fuel mixture is obtained and combustion is completed, and the gases forced out into the main cylinder where they expand, forcing the piston downward. The compression pressure is about 550 lb. per square inch, and the peak pressure at combustion is from 600 to 750 lb. per square inch. With increased engine speeds increased turbulence is effected inside the combustion chamber, hence the efficiency of combustion is not decreased.

The same Bosch fuel pump and governing system as described for the Hesselman engine is used. A different type of injection valve, the

Bosch hydraulically operated, spring-loaded valve is used. The needle valve is opened by oil pressure exerted upward on the lower ring, lifting the valve against the force of the spring, which is adjustable. The nozzle has only one orifice, $1/16$ in. or slightly larger in diameter, but the atomization is effective due to the "pintle" tip on the needle valve. This type of spray nozzle practically eliminates clogging with dirt or carbon.

As an aid in starting in cold weather, an electric glow plug, resembling a spark plug, is installed, projecting into the combustion chamber. This is connected in the path of return current flow in the starting motor circuit. The glow plug adds heat to the starting heat of compression while the starting motor is in service. At other times it is inactive.

334. Discussion of the Solid-injection System.—With *solid* or *mechanical injection,* the use of air as a propellant for the fuel is eliminated, and a fuel pump alone is used to force the oil into the combustion chamber of the engine. Solid injection is used by high-compression ignition oil engines and the hot-surface ignition engines operating on low compression. The development of the first type of engine has resulted in elimination of costly and troublesome air-compression equipment. At the same time there is an attempt to attain the almost perfect fuel atomization that is incident with air injection. In the solid-injection nozzle the valve is of simple design. It is held tight against a conical seat at the end of the nozzle by a spring which is, to a certain extent, adjustable. The fuel pump, in operation, exerts a pressure which overcomes the spring to a point where the valve is opened and injection occurs. When the pressure drops the spring closes the valve.

Such a valve introduces the fuel in the liquid form, and the atomization is somewhat affected by the design of the combustion chamber. The usual design is to provide an auxiliary chamber in the cylinder head, known as the precombustion chamber. It is connected with the cylinder by one large or many small ports, depending on the design. In this chamber the lighter fractions of the oil ignite first, burning instantaneously with the effect of an explosion, and as a result of the extreme pressure the heavier portion of the oil is swept into the cylinder with a fair degree of atomization. The indicator diagram from such an engine resembles, very closely, that of an engine operating on the Otto cycle.

335. Miscellaneous Oil Engines.—One of the early solid-injection engines embodied the so-called Hvid cup. This cup forms the precombustion chamber, and is arranged with ports to the fuel valve and to

the cylinder. At the beginning of the compression stroke, fuel is forced into the cup, and, near the end of the stroke, the cup is automatically opened to the cylinder, and the action briefly described above takes place. The Dodge oil engine is one of many that use the Hvid principle, or some modification of it.

Another design utilizing this type of ignition is the Price engine. The precombustion chamber of this engine is provided with fuel jets which collide amid a violent turbulence of air. The precombustion chamber is located in the cylinder head and has a large opening to the cylinder. The two fuel-injection nozzles are placed at opposite points in this chamber. Most of the high-compression oil engines using solid

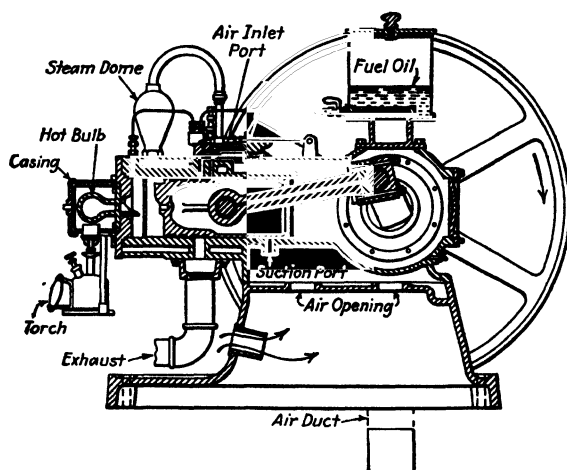



FIG. 336.—Mietz and Weiss hot-surface ignition engine.

injection utilize similar principles of obtaining ignition, their chief differences being in the shape or location of the combustion chamber.

Hot-surface type of *ignition* is older than the Diesel engine, having been first used in the Hornsby-Ackroyd engine which antedated Dr. Diesel's first practical engine. The Hornsby-Ackroyd engine had an uncooled precombustion chamber in the cylinder head. Due to the heat of continuous combustion, the walls of the chamber became hot enough to effect combustion of the previously injected fuel at the end of the compression stroke. In contact with the hot surface, the fuel was ignited and burned explosively, practically all of the combustion occurring in the hot chamber. The burned gases issued against the piston, and the maximum pressure seldom exceeded 200 lb. per square inch. The operation was similar to the Otto cycle.

An interesting development and an application of hot-surface ignition still in use is found in the Mietz and Weiss engine, illustrated in

Fig. 336. This was the first type of American, two-stroke cycle, low-compression, explosion oil engine. It has many interesting features aside from the hot-surface ignition. A small plunger pump, operated by an eccentric on the crank shaft, forces a charge of fuel into the cylinder, discharging it on a projecting lip of the hot bulb. This occurs at the time the piston, on the compression stroke, has just covered the exhaust port. In theory, the hot surface partially vaporizes the fuel, and during compression this vapor is forced into the hot-bulb chamber. Contact with the hot surface causes an explosion, thus producing great turbulence in the cylinder as the fuel burns, which is during the start of the power stroke of the piston. A centrifugal governor mounted around the shaft changes the eccentricity of the fuel-pump eccentric, thereby altering the stroke of the plunger and varying the amount of fuel supplied for each stroke according to the load. Cooling of the cylinder walls is accomplished by a "dead-end" water jacket. In a water chamber adjacent and connected to the cylinder jacket is a float-controlled valve, which maintains the water in the jackets at constant level. The water is not circulated, and the jacket generates steam when the engine becomes hot. The water vaporizes and the vapor passes through the steam dome and a bent tube to the air-intake port of the cylinder, after which it enters the cylinder along with the scavenging air. This has the effect of moderating the cylinder temperature and permits higher compression pressures than would otherwise be practical. For heavy loads, water may also be admitted directly to the intake ports which more effectively eliminates excessive cylinder temperatures and preignition.

336. Fuel Pumps and Governors.—Diesel engines using the closed-type, air-injection nozzle require a valve of sturdy construction, as its service is severe. The fuel pump discharges against injection air pressures of approximately 1,000 lb. per square inch. Governors are often designed, not to alter the stroke of the pump plunger, but to control the weight of each charge of fuel by regulating the time of opening of the valves.

Figure 337 shows, diagrammatically, the operating principle of a constant-stroke fuel pump and also the method of governing. The pump plunger *a*, and the suction-valve plunger *b* have constant strokes and are driven by eccentrics on a vertical shaft. During the suction stroke of the pump plunger, the suction valve is opened by plunger *b* by bell crank *c*, and fuel flows into the pump chamber. As the pump plunger reverses, on the discharge stroke, the suction valve remains open, and, during the early part of this stroke, fuel is returned to the suction line. At a certain point the suction-valve plunger

moves away from the bell crank, and the suction valve closes. During the remainder of the discharge stroke fuel flows through the check valve and through the delivery line to the engine cylinder. The

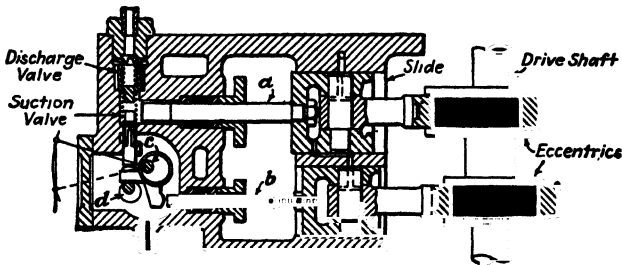



FIG. 337.—Busch-Sulzer fuel pump.

axis about which the bell crank turns is under the control of the governor and is changed as the engine speed varies. With a decrease in load and consequent increase in engine speed, the bell-crank position is automatically changed so that the regulating rod, earlier in its stroke, makes contact with the bell crank and increases the lift of the suction valve. The closing of this valve occurs later. During the discharge stroke, the delivery check valve does not open until the suction valve is closed, and as long as the suction valve is open, fuel is returned to the suction chamber. The small eccentric *d* can be operated by hand and its rotation holds the suction valve off its seat, which stops the delivery of fuel to the engine. A further rotation of the eccentric causes both the discharge and suction valve to open for the purpose of removing air from the fuel lines.

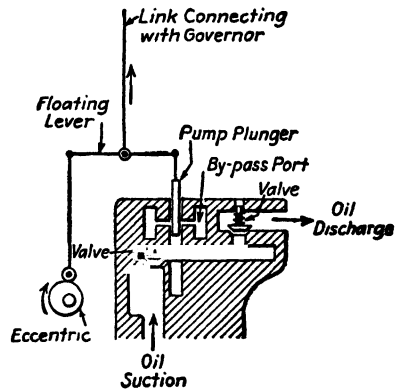



FIG. 338.—Diagrammatic sketch of a constant-stroke fuel pump.

Another type of pump-governing device often used has a constant stroke plunger, which supplies the same quantity of fuel each stroke. The fuel is discharged into the cylinder feed line until a by-pass valve is opened by the governor. The opening of this valve stops the flow to the cylinder by returning fuel to a suction chamber.

The operating principle of a commonly used type of constant-stroke fuel pump is illustrated in Fig. 338. The plunger receives its motion from an eccentric cam and linkage, as shown. The position

TABLE 15-5 — TYPICAL DIESEL PLANTS, SIZES OVER 200 HP.

Owner	Location	Service	Equipment driven	Builder	Number of units	Hp each	R p m	Number of cylinders	Cylinder bore	Stroke	Cycle	Starting	Fuel injection	Fuel consumption, lb per hp-hr
Freeport	New York	Mun	Alt	Bu S	1	3 000	240	10	19½	27	2	A	M	0 40
Rockville Center	New York	Mun	Alt	McI	1	2 865	120	8	29	48	4	A	A	0 40
Ill Central R R	Chicago, Ill	Loco	Gen	Bu S	1	2 000	550	10	14	16	2	E	M	0 41
U S District Engineer	Louisville, Ky	Dredge	Pump	Bu S	1	1 700	250	6	19½	27	2	A	M	0 40
			Gen		1	840	250	6	15½	21	2	A	M	0 40
			Prop		2	750	250	6	15½	21	2	A	M	0 40
Russel	Kansas	Lt & Pwr	Gen	D L V	1	1 600	200	8	22	30	4	A	M	0 41
Longmont	Colorado	Mun	Gen	F M	1	1 400	300	8	16	20	2	A	M	0 39
Marshall	Michigan	Mun	Alt	Nor	1	1 250	257	5	17	25	4	A	A	
Harlan	Iowa	Mun	Gen	Ful	1	1 000	225	8	17½	24½	4	A	A	
Eastport	Maine	Pwr	Alt	Win	2	700	360	8	14	16	4	A	M	0 39
N Y, N H & H R R	New Haven Conn	Loco	Gen	C B	5	660	750	8	10½	12	4	A	M	0 40
Baldwin Loco Co	Eddystone, Pa	Pwr	Loco	D L V	1	660	600	6	12½	15½	4	A	M	0 42
Namm store	Brooklyn, N Y	Pwr & Lt	Alt & comp	Wor	4	300	514	6	10¾	14½	4	A	M	0 39
		Air cond	Alt	Wor	1	150	514	6	8	10½	4	A	M	0 39
Bowling Green Mill	Bowling Green Ky	Flour	Mill	Buc	1	225	400	6	9½	14	4	A	M	0 40
8th Ave & 50th St Bldg	New York N Y	Pwr	Gen	I R	1	225	360	4	11	18	4	A	M	

NOTE — Abbreviations Buc = Buckeye, Bu S = Busch Sulzer C B = Cooper Bessemer, D L V = De La Vergne, F M = Fairbanks Morse, Ful = Fulton, I R = Ingersoll-Rand McI = McIntosh & Seymour Nor = Nordberg Win = Winton Wor = Worthington

Alt = alternator, Comp = compressor, Gen = generator, Lt = light, Mun = municipal, Prop = propeller, Pwr = power

Fuel injection and starting A = air E = electrical M = mechanical

TABLE 15-6 —DATA FOR VARIOUS MAKES OF INTERNAL-COMBUSTION ENGINES[1]

Maker and engine	No cylinders	Size, in	R p m	Piston speed, ft per min	Brake m e p	Brake hp	Weight, lb.	Weight per hp, lb	Ratio weight to cu in. displacement
1 De La Vergne, heavy-duty Diesel	6	17 by 24	225	900	65	600	145,000	242	4 45
2 Atlas Imperial, motor-boat drive	6	8½ by 12	325	650	75	125	17,500	140	4 30
3. Waukesha, industrial, gasoline	6	7¾ by 8½	1 000	1 415	91	275	7,340	26 6	3 10
4 Winton, marine, Diesel	6	8 by 10	700	1 170	75	200	8,000	40	2 67
5 Buda, gasoline truck	6	4 by 5⅛	2,000	1,700	75	73	946	13	2 45
6 Continental, auto, gasoline	6	3⅜ by 4	2,900	1 930	78	61	468	7 69	2 18
7. Beardmore, Diesel, locomotive	6	8¼ by 12	750	1,500	83	300	4 000	13 3	1 04
8 Curtiss, aero, gasoline	12	5⅛ by 6¼	2,400	2,500	128	600	755	1 25	0 49

TABLE 15-7 —HIGH-SPEED DIESEL ENGINES[1]

Maker and engine	No cylinders	Size, in	R p m	Piston speed, ft per min	Brake m e p	Brake hp	Weight, lb.	Weight per hp, lb	Ratio weight to cu in. displacement
1 Buda	4	6 by 8	1,000	1 330	80	92	3,100	33	3 45
2 M. A. N	6	4 7 by 7 1	1,000	1,180	72	68	1,950	29	2 6
3 Linke-Hofmann	4	4 5 by 6 5	1,200	1 300	80	50	1,100	22	2 7
4. Deutz..	6	4 5 by 6 7	1,000	1,115	74	60	1,600	27	2 5
5. Koerting	6	5 1 by 7 1	1,200	1,420	67 5	90	3,300	36	3 75
6. Mercedes-Benz	6	4 1 by 6 5	1,300	1,400	83	70	1,600	23	3 1
7. Maybach	6	5 5 by 7 1	1,300	1,535	90	150	2,650	17 7	2 6
8. Junkers	2	3 15 by 11 8	1,000	983	86	40	640	16	3 5

[1] *Transactions*, A S M.E., vol 51, No 31, Otto Nonnenbruch, I P Morris and De La Vergne, Inc

of the floating lever is controlled by the governor, and the amount of fuel delivered to the cylinder feed line depends upon the distance, past the by-pass port, that the plunger moves on its down stroke. The position of the pivot of the floating lever is determined by the governor, which affects the piston travel and thus regulates the fuel delivered per stroke. This type of governor mechanism is used in conjunction with the open type of air-injection nozzle.

There is also a *variable-stroke* type of fuel pump, which is driven by an eccentric, the eccentricity of which is changed by the governor. As the load decreases, the throw of the eccentric is made less, thus reducing the amount of fuel metered to the cylinder.

Multiple-cylinder oil engines, using air injection, are generally designed with one pump plunger for each engine cylinder. Adjustments for balancing the power distribution among the cylinders can be made in the fuel-feed mechanism.

Problems

1. Compute the Carnot cycle efficiencies for the following temperatures at the start of isothermal expansion: 300, 600, 1000, 2000, and 3000°F. Refrigerator temperature 200°F. Plot efficiencies against initial temperature.

2. Compute the Carnot cycle efficiencies for the following refrigerator temperatures: 0, 32, 150, 212, and 300°F. Initial temperature constant at 750°F. Plot efficiencies against refrigerator temperature.

3. Referring to the Carnot cycle diagram (Fig. 296), the refrigerator temperature is 140°F.; pressure at C is 15 lb. per square inch absolute; volume at A is 10 per cent of the volume at C ($V_c = 12.5$ cu. ft.). The cylinder contains air, and the heat rejected per cycle is 50 B.t.u. Determine the work done per cycle in foot-pounds.

4. Referring to the Diesel cycle (Fig. 297), the clearance is 10 per cent of the displacement. Calculate the cycle efficiency, if combustion takes the following percentages of the stroke 3, 6, 9, 12, 15 and 20.

5. Plot efficiency against the temperature after combustion, (t_c), from the results of Problem 4, if t_b is assumed to be 1250°F. (Fig. 297).

6. A Diesel cycle shows that fuel is injected for 8 per cent of the stroke. What are the values of efficiency for the following clearance volumes, expressed as percentages of the stroke: 3, 9, 15, 21, 27 and 33.

7. Plot efficiency against compression ratio for the results of Problem 6.

8. The conditions at the start of compression in the theoretical Diesel cycle are 14 lb. per square inch absolute; 150°F.; a total volume of 5.4 cu. ft. The clearance volume is 17.5 per cent of the displacement, and the heat added during the cycle is 125 B.t.u. If the cycle is repeated 200 times per minute, what is the horsepower developed?

9. Determine the efficiency of the Otto cycle for compression ratios of 5, 10, 15, 20, 25 and 30.

10. Plot efficiency against compression ratio from the results of Problem 9.

11. In the Otto cycle $w = 0.5$ lb. of air; the pressure at the start of compression is 14.7 lb. per square inch absolute; temperature 85°F.; the pressure after combus-

tion is 375 lb. per square inch absolute, and the temperature 2650°F. Calculate the efficiency and clearance as a percentage of displacement.

12. Determine the weight of air required for the combustion of 1 lb. of natural gas with the following volumetric analysis: H_2, 2.5 per cent; CH_4, 90.5; C_2H_4, 0.5; O_2, 0.7; N_2, 5.8.

13. Determine the theoretical volume of air required per cubic foot of the gas of Problem 12.

14. A coke-oven gas burns with 50 per cent excess air. Determine the volume of each product of combustion of 1 cu. ft. of the gas. Volumetric analysis as follows: H_2, 45.8 per cent; CO, 9.2; CH_4, 32.4; C_2H_4, 1.2; O_2, 6.0; N_2, 5.4.

15. A blast-furnace gas burns with 100 per cent of excess air. Determine the weight of each product of combustion per pound of the gas. The volumetric analysis follows: H_2, 3 per cent; CO, 25; CO_2, 10; N_2, 62.

16. Using the universal gas constant, determine the value of R for H_2, N_2, O_2, CO_2, CO.

17. Determine the theoretical weight of air required to burn 1 lb. of gasoline, (C_8H_{18}).

18. Determine the theoretical weight of the products of combustion from burning 1 mol. of kerosene ($C_{12}H_{26}$).

19. Determine the lower heating value per cubic foot of the gas of Problem 15 if the temperature and pressure are 150°F. and 20 lb. per square inch absolute, respectively.

20. Determine the higher heating value of 1 mol. of the gas of Problem 14 at 14.7 lb. per square inch absolute, and 68°F.

21. Calculate indicated horsepower for the single-acting engines listed in the following table:

Size, in.	No. cylinders	Cycle	Indicated m.e.p.	R.p.m.	Brake load	Brake arm
a. 8½ by 11	4	4	68	277	250	3
b. 13½ by 17	3	4	75	155	800	3
c. 10½ by 15	4	4	75	300	400	5
d. 16 by 20	7	2	43	260	1,400	8
e. 23½ by 27½	3	2	65	180	2,800	8
f. 17 by 23	3	2	55	257	1,250	6.5
g. 11 by 15	6	4	72	360	670	5

22. Calculate the brake horsepower of each engine in Problem 21.

23. For Problem 21, calculate the brake mean effective pressures of each engine.

24. Determine the torque of each engine in Problem 21.

25. The following table lists results from Diesel-engine operation. Determine the thermal efficiency of each engine.

No.	B.hp.	Fuel, °Bé.	Fuel B.t.u. per gal.	Output, kw.-hr. per gal.	Cooling water, °F.		Exhaust gases, °F.	Cooling water, gal. per min.
					In	Out		
a.	1,250	18	147,000	11 4	70	110	600	39
b.	2,200	18	147,000	10 0	90	130	750	54
c.	400	25	143,000	10.8	80	120	600	19
d.	560	24	144,000	8 65	80	115	650	23
e.	900	20	147,000	11 2	70	130	750	30
f.	600	29	148,500	10.5	65	115	600	25
g.	560	25	148,618	10 37	65	100	625	21

26. With a room temperature of 75°F., and an air-fuel ratio by weight of 13: 1, calculate a heat balance for the engines of Problem 25 (Use specific heat of exhaust gases = 0.246.)

27. A six-cylinder, two-stroke cycle, single-acting Diesel engine, rated at 3,750 b.hp., cylinder size 29 by 44 in., gave the following data on acceptance tests. The fuel had a heating value of 18,738 B.t.u. per pound, Baumé gravity 20 6.

	Test 1	Test 2	Test 3	Test 4	Test 5
Fuel oil, lb. per hr .	584 0	871 0	1,218 8	1,585 6	1,728 0
Compressor, i.hp .	171 6	212 4	236 3	250 2	249 9
Scavenging pump, i hp .	387 5	385 0	376 2	386 0	395 0
Indicated m.e p	37 28	57 1	71 45	89 57	97 4
Mechanical efficiency	45 8	59 9	71 6	76 3	77 2
R.p.m	125 65	125 1	125 17	124 71	125 0
Guaranteed fuel consumption, lb per b.hp-hr	0 72	0 54	0 48	0 45	0 45

Calculate for each test (*a*) the fuel consumption, pounds per horsepower-hour, brake and indicated, and (*b*) the percentage of rated load. Plot curves of the guaranteed and actual economy results.

28. Calculate the air-standard efficiency of the ideal dual cycle for the following conditions: At start of compression $P = 13.7$ lb. abs ; $t = 600$°F. abs ; volume 1 cu. ft.; at end of compression, $p = 100$ lb. abs.; maximum pressure, 250 lb. abs.; volume at end of constant-pressure combustion, 0.4 cu ft.

29. An ideal engine, on the Carnot cycle, receives 200 B.t.u. per minute; temperature, source 1500°F., refrigerator 50°F. Calculate the engine horsepower.

30. In an ideal engine on the Otto cycle (Fig 298) the following conditions are known: $p_a = 15$ lb. per square inch absolute; $t_a = 200$°F.; $t_b = 2240$°F.; per cent clearance = 12.5. Calculate the cycle efficiency, and p_c.

31. Solve Problem 30 for an engine on the Diesel cycle.

32. An ideal engine, on the theoretical Diesel cycle (Fig. 297), develops 100 hp. at a speed of 200 cycles per minute. The following data are known: $p_a = 14.2$ lb. per square inch absolute; $t_a = 160$°F.; $v_a = 2.8$ cu. ft.; per cent clearance = 15. Calculate Q_1, the heat absorbed per cycle.

33. Solve Problem 32 for an engine on the Otto cycle.

34. Volumetric analysis of a fuel gas: C_2H_6 35; C_4H_{10} 40; CO_2 10; N_2 15 per cent. For theoretical combustion calculate:

a. Volume of air required per cubic foot of fuel (14.7 lb. abs. and 32°F.).

b. Volume of combustion products per cubic foot of fuel (14.7 lb. abs. and 32°F.).

c. Weight of air required per pound of fuel.

d. Weight of combustion products per pound of fuel.

e. Volume of combustion products per pound of fuel (14.7 lb. abs. and 32°F.).

f. Weight of combustion products per cubic foot of fuel (14.7 lb. abs. and 32°F.).

g. Higher heating value per cubic foot (100 lb. abs. and 300°F.).

h. Higher heating value per pound (100 lb. abs. and 300°F.).

i. Value of R for the fuel.

35. Single-acting, Diesel four-stroke cycle, 14 by 18 in., four cylinder, average indicated m.e.p. 57; 400 r.p.m.; Prony brake arm 5 ft., net load 750 lb.; fuel 145,000 B.t.u. per gallon, 24° Bé., output 13.5 hp.-hr. per gallon. Calculate:

a. Brake horsepower.

b. Indicated horsepower.

c. Torque.

d. Actual thermal efficiency.

e. Brake m.e.p. (use brake horsepower = 280).

f. Higher heating value of fuel at 70°F., B.t.u. per pound.

36. Volumetric analysis of a fuel gas: C_2H_2 26, C_3H_8 44, N_2 18, CO_2 12 per cent. If this gas burns completely with 30 per cent excess air, calculate:

a. Volume of air supplied per cubic foot of fuel (14.7 lb. abs. and 32°F.).

b. Weight of combustion products per pound of fuel.

c. Higher heating value per pound of fuel.

d. Value of R for the fuel.

e. Volume of combustion products per cubic foot of fuel (14.7 lb. abs. and 32°F.).

f. Weight of combustion products per cubic foot of fuel (14.7 lb. abs. and 32°F.).

g. Volume of combustion products per pound of fuel (14.7 lb. abs. and 32°F.).

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APPENDIX

TABLES OF PROPERTIES OF VAPORS

TABLE 1.—SATURATED STEAM: TEMPERATURE TABLE[1]

Temp. Fahr.	Abs. Press. Lb./Sq. In.	Specific Volume: Sat. Liquid	Specific Volume: Evap.	Specific Volume: Sat. Vapor	Total Heat: Sat. Liquid	Total Heat: Evap.	Total Heat: Sat. Vapor	Entropy: Sat. Liquid	Entropy: Evap.	Entropy: Sat. Vapor	Temp. Fahr.
t	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
32°	0.0887	0.01602	3301	3301	0.00	1073.4	1073.4	0.0000	2.1834	2.1834	32°
34	0.0961	0.01602	3059	3059	2.01	1072.3	1074.3	0.0041	2.1723	2.1764	34
36	0.1041	0.01602	2835	2835	4.03	1071.2	1075.3	0.0081	2.1614	2.1695	36
38	0.1126	0.01602	2632	2632	6.04	1070.2	1076.2	0.0122	2.1505	2.1627	38
40°	0.1217	0.01602	2445	2445	8.05	1069.1	1077.1	0.0162	2.1397	2.1559	40°
42	0.1315	0.01602	2272	2272	10.05	1068.0	1078.0	0.0202	2.1290	2.1492	42
44	0.1420	0.01602	2112	2112	12.05	1066.9	1078.9	0.0242	2.1184	2.1426	44
46	0.1532	0.01602	1965.3	1965.3	14.06	1065.8	1079.8	0.0282	2.1078	2.1360	46
48	0.1652	0.01602	1829.7	1829.7	16.06	1064.7	1080.8	0.0321	2.0974	2.1295	48
50°	0.1780	0.01602	1704.8	1704.8	18.06	1063.6	1081.7	0.0361	2.0870	2.1231	50°
52	0.1918	0.01602	1588.3	1588.3	20.06	1062.5	1082.6	0.0400	2.0767	2.1167	52
54	0.2063	0.01602	1482.4	1482.4	22.06	1061.4	1083.5	0.0439	2.0665	2.1104	54
56	0.2219	0.01603	1383.5	1383.5	24.05	1060.3	1084.4	0.0478	2.0564	2.1042	56
58	0.2384	0.01603	1292.8	1292.8	26.05	1059.3	1085.3	0.0517	2.0464	2.0980	58
60°	0.2561	0.01603	1208.0	1208.0	28.05	1058.2	1086.2	0.0555	2.0364	2.0919	60°
62	0.2749	0.01603	1129.7	1129.7	30.05	1057.1	1087.1	0.0594	2.0265	2.0859	62
64	0.2949	0.01604	1057.1	1057.1	32.04	1056.0	1088.0	0.0632	2.0167	2.0799	64
66	0.3162	0.01604	989.7	989.6	34.04	1054.9	1089.0	0.0670	2.0069	2.0739	66
68	0.3388	0.01605	927.1	927.1	36.03	1053.8	1089.8	0.0708	1.9973	2.0680	68
70°	0.3628	0.01605	869.0	869.0	38.03	1052.7	1090.8	0.0746	1.9877	2.0622	70°
72	0.3883	0.01606	814.9	815.0	40.02	1051.6	1091.7	0.0783	1.9782	2.0565	72
74	0.4153	0.01606	764.8	764.8	42.02	1050.6	1092.6	0.0820	1.9687	2.0507	74
76	0.4440	0.01606	718.0	718.0	44.01	1049.5	1093.5	0.0858	1.9593	2.0451	76
78	0.4744	0.01607	674.5	674.5	46.00	1048.4	1094.4	0.0895	1.9500	2.0395	78
80°	0.5067	0.01607	633.8	633.8	48.00	1047.3	1095.3	0.0932	1.9407	2.0340	80°
82	0.5409	0.01608	595.9	595.9	50.00	1046.2	1096.2	0.0969	1.9316	2.0285	82
84	0.5772	0.01608	560.5	560.5	52.00	1045.1	1097.1	0.1006	1.9224	2.0230	84
86	0.6153	0.01609	527.6	527.7	54.00	1044.0	1098.0	0.1042	1.9134	2.0176	86
88	0.6555	0.01609	497.1	497.1	56.00	1042.9	1098.9	0.1079	1.9044	2.0123	88
90°	0.6980	0.01610	468.5	468.5	58.00	1041.8	1099.8	0.1115	1.8955	2.0070	90°
92	0.7429	0.01610	441.7	441.8	59.98	1040.7	1100.7	0.1152	1.8866	2.0018	92
94	0.7902	0.01611	416.8	416.8	61.97	1039.6	1101.6	0.1188	1.8778	1.9966	94
96	0.8403	0.01612	393.3	393.3	63.96	1038.5	1102.5	0.1224	1.8691	1.9915	96
98	0.8930	0.01612	371.4	371.4	65.94	1037.4	1103.4	0.1260	1.8604	1.9864	98
100°	0.9487	0.01613	350.8	350.8	67.93	1036.3	1104.2	0.1295	1.8518	1.9813	100°
105	1.1009	0.01615	305.0	305.0	72.91	1033.5	1106.4	0.1384	1.8305	1.9689	105
110	1.274	0.01616	265.8	265.8	77.89	1030.8	1108.6	0.1472	1.8095	1.9567	110
115	1.470	0.01618	232.3	232.3	82.89	1027.9	1110.8	0.1559	1.7889	1.9448	115
120°	1.692	0.01620	203.5	203.6	87.88	1025.1	1113.0	0.1646	1.7685	1.9331	120°
125	1.941	0.01622	178.9	178.9	92.87	1022.2	1115.1	0.1731	1.7485	1.9216	125
130	2.221	0.01625	157.62	157.64	97.86	1019.4	1117.2	0.1816	1.7288	1.9104	130
135	2.536	0.01627	139.15	139.17	102.85	1016.5	1119.3	0.1901	1.7094	1.8995	135
140	2.887	0.01629	123.20	123.22	107.84	1013.6	1121.4	0.1984	1.6903	1.8887	140
145°	3.280	0.01632	109.29	109.31	112.84	1010.6	1123.5	0.2067	1.6715	1.8782	145°
150	3.716	0.01634	97.21	97.23	117.84	1007.7	1125.5	0.2149	1.6530	1.8679	150
155	4.201	0.01637	86.64	86.66	122.85	1004.7	1127.6	0.2231	1.6347	1.8578	155
160	4.739	0.01639	77.38	77.40	127.85	1001.8	1129.6	0.2312	1.6168	1.8479	160
165	5.334	0.01642	69.26	69.28	132.85	998.8	1131.7	0.2392	1.5991	1.8383	165
170°	5.990	0.01645	62.13	62.14	137.85	995.8	1133.7	0.2472	1.5816	1.8288	170°
175	6.716	0.01648	55.81	55.82	142.86	992.8	1135.7	0.2551	1.5644	1.8195	175
180	7.510	0.01650	50.26	50.28	147.87	989.8	1137.7	0.2629	1.5475	1.8105	180
185	8.382	0.01654	45.35	45.36	152.87	986.8	1139.7	0.2707	1.5308	1.8016	185
190	9.336	0.01656	40.99	41.01	157.89	983.8	1141.7	0.2785	1.5144	1.7929	190
195°	10.385	0.01660	37.10	37.12	162.91	980.8	1143.7	0.2862	1.4982	1.7844	195°
200	11.525	0.01663	33.65	33.67	167.94	977.7	1145.6	0.2938	1.4822	1.7760	200
205	12.769	0.01665	30.57	30.59	172.97	974.6	1147.6	0.3014	1.4664	1.7678	205
210	14.123	0.01669	27.82	27.83	177.99	971.5	1149.5	0.3089	1.4508	1.7597	210
212	14.696	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	212
215°	15.591	0.01673	25.35	25.37	183.02	968.3	1151.3	0.3164	1.4353	1.7517	215°
220	17.188	0.01676	23.14	23.16	188.06	965.1	1153.1	0.3238	1.4200	1.7439	220
225	18.915	0.01680	21.15	21.17	193.09	961.8	1154.9	0.3313	1.4049	1.7362	225
230	20.78	0.01683	19.371	19.388	198.15	958.6	1156.7	0.3386	1.3900	1.7286	230
235	22.80	0.01687	17.761	17.778	203.21	955.3	1158.5	0.3460	1.3752	1.7212	235

[1] MOYER, CALDERWOOD and POTTER, "Principles of Engineering Thermodynamics."

Mollier Diagram'

TABLE 1.—SATURATED STEAM: TEMPERATURE TABLE.[1]—(*Continued*)

Temp. Fahr.	Abs. Press. Lb./Sq. In.	Specific Volume Sat. Liquid	Specific Volume Evap.	Specific Volume Sat. Vapor	Total Heat Sat. Liquid	Total Heat Evap.	Total Heat Sat. Vapor	Entropy Sat. Liquid	Entropy Evap.	Entropy Sat. Vapor	Temp. Fahr.
t	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
240°	24.97	0.01690	16.307	16.324	208.26	952.0	1160.2	0.3532	1.3607	1.7138	240°
245	27.31	0.01695	14.991	15.008	213.33	948.6	1161.9	0.3604	1.3462	1.7066	245
250	29.82	0.01698	13.807	13.824	218.39	945.2	1163.6	0.3675	1.3320	1.6995	250
255	32.53	0.01702	12.726	12.743	223.47	941.8	1165.3	0.3746	1.3179	1.6925	255
260	35.43	0.01706	11.745	11.762	228.55	938.4	1166.9	0.3817	1.3040	1.6856	260
265°	38.54	0.01710	10.854	10.871	233.65	934.9	1168.6	0.3888	1.2901	1.6789	265°
270	41.85	0.01714	10.044	10.061	238.74	931.4	1170.1	0.3958	1.2765	1.6723	270
275	45.40	0.01719	9.304	9.321	243.84	927.8	1171.6	0.4028	1.2630	1.6658	275
280	49.20	0.01723	8.626	8.644	248.95	924.2	1173.2	0.4097	1.2496	1.6593	280
285	53.25	0.01728	8.007	8.024	254.08	920.6	1174.7	0.4166	1.2363	1.6529	285
290°	57.55	0.01732	7.442	7.459	259.20	917.0	1176.2	0.4234	1.2233	1.6467	290°
295	62.13	0.01737	6.923	6.940	264.34	913.4	1177.7	0.4303	1.2103	1.6406	295
300	67.01	0.01742	6.446	6.464	269.48	909.6	1179.1	0.4370	1.1974	1.6345	300
305	72.18	0.01747	6.009	6.026	274.64	905.8	1180.4	0.4438	1.1847	1.6285	305
310	77.68	0.01752	5.606	5.623	279.80	902.1	1181.9	0.4505	1.1721	1.6226	310
315	83.50	0.01757	5.234	5.252	284.98	898.2	1183.2	0.4571	1.1596	1.6167	315
320°	89.65	0.01762	4.892	4.910	290.17	894.4	1184.6	0.4638	1.1472	1.6110	320°
325	96.16	0.01768	4.573	4.591	295.38	890.5	1185.9	0.4705	1.1349	1.6054	325
330	103.03	0.01773	4.285	4.303	300.59	886.5	1187.1	0.4770	1.1227	1.5998	330
335	110.31	0.01779	4.016	4.034	305.81	882.5	1188.3	0.4836	1.1106	1.5942	335
340	117.99	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	340
345°	126.10	0.01791	3.535	3.553	316.30	874.3	1190.6	0.4966	1.0866	1.5832	345°
350	134.62	0.01797	3.321	3.338	321.55	870.2	1191.8	0.5031	1.0748	1.5779	350
355	143.58	0.01804	3.121	3.139	326.82	866.0	1192.8	0.5095	1.0630	1.5725	355
360	153.01	0.01810	2.936	2.954	332.10	861.7	1193.8	0.5160	1.0514	1.5673	360
365	162.93	0.01817	2.764	2.782	337.40	857.4	1194.8	0.5224	1.0397	1.5621	365
370°	173.33	0.01823	2.604	2.622	342.71	853.0	1195.7	0.5288	1.0282	1.5570	370°
375	184.23	0.01830	2.455	2.473	348.05	848.6	1196.7	0.5352	1.0167	1.5519	375
380	195.70	0.01836	2.315	2.333	353.39	844.1	1197.5	0.5415	1.0053	1.5468	380
385	207.71	0.01844	2.185	2.203	358.76	839.5	1198.3	0.5479	0.9939	1.5418	385
390	220.29	0.01850	2.0631	2.0816	364.14	834.9	1199.0	0.5542	0.9826	1.5368	390
395	233.47	0.01858	1.9489	1.9675	369.54	830.3	1199.8	0.5605	0.9714	1.5319	395
400°	247.25	0.01865	1.8421	1.8608	374.96	825.5	1200.4	0.5668	0.9602	1.5270	400°
410	276.72	0.01880	1.6493	1.6681	385.86	815.8	1201.6	0.5792	0.9381	1.5173	410
420	308.82	0.01896	1.4792	1.4982	396.84	805.8	1202.7	0.5916	0.9161	1.5077	420
430	343.71	0.01911	1.3295	1.3486	407.91	795.5	1203.4	0.6039	0.8942	1.4982	430
440	381.59	0.01928	1.1965	1.2158	419.07	784.9	1203.9	0.6162	0.8724	1.4887	440
450°	422.61	0.0195	1.0782	1.0977	430.3	773.8	1204.1	0.6284	0.8507	1.4792	450°
460	466.94	0.0196	0.9730	0.9927	441.7	762.3	1204.0	0.6407	0.8290	1.4696	460
470	514.76	0.0198	0.8793	0.8991	453.2	750.3	1203.5	0.6530	0.8071	1.4601	470
480	566.26	0.0200	0.7951	0.8151	465.0	737.8	1202.8	0.6654	0.7852	1.4506	480
490	621.67	0.0202	0.7195	0.7398	477.0	724.7	1201.7	0.6779	0.7632	1.4410	490
500°	681.09	0.0205	0.6516	0.6721	489.1	711.1	1200.2	0.6904	0.7410	1.4314	500°
510	744.74	0.0207	0.5903	0.6110	501.6	696.9	1198.4	0.7031	0.7187	1.4218	510
520	812.72	0.0210	0.5347	0.5557	514.2	682.1	1196.3	0.7158	0.6963	1.4121	520
530	885.31	0.0213	0.4845	0.5058	527.0	666.8	1193.8	0.7286	0.6738	1.4024	530
540	962.73	0.0216	0.4394	0.4610	540.0	651.0	1191.0	0.7414	0.6512	1.3926	540
550°	1045.4	0.0219	0.3982	0.4201	553.2	634.5	1187.8	0.7543	0.6285	1.3828	550°
560	1133.4	0.0223	0.3605	0.3828	566.7	617.5	1184.2	0.7672	0.6056	1.3728	560
570	1227.6	0.0227	0.3261	0.3488	580.4	599.7	1180.2	0.7802	0.5825	1.3626	570
580	1327.2	0.0231	0.2949	0.3180	594.4	581.3	1175.7	0.7932	0.5592	1.3524	580
590	1432.7	0.0236	0.2664	0.2900	608.7	562.2	1170.8	0.8064	0.5356	1.3420	590
600°	1544.6	0.0241	0.2401	0.2642	623.2	542.3	1165.5	0.8198	0.5118	1.3316	600°
610	1663.2	0.0247	0.2159	0.2406	638.0	521.4	1159.5	0.8332	0.4875	1.3208	610
620	1788.8	0.0254	0.1933	0.2186	653.4	499.2	1152.5	0.8470	0.4623	1.3093	620
630	1921.9	0.0261	0.1721	0.1982	669.5	474.8	1144.3	0.8612	0.4358	1.2970	630
640	2062.8	0.0269	0.1522	0.1791	686.6	447.9	1134.5	0.8763	0.4073	1.2836	640
650°	2211.4	0.0278	0.1331	0.1610	705.2	417.7	1122.8	0.8924	0.3764	1.2688	650°
660	2368.6	0.0290	0.1148	0.1437	725.3	383.6	1108.8	0.9097	0.3426	1.2523	660
670	2534.2	0.0304	0.0966	0.1269	747.5	344.4	1091.9	0.9287	0.3049	1.2336	670
680	2709.7	0.0322	0.0781	0.1102	772.6	298.5	1071.2	0.9499	0.2619	1.2119	680
690	2896.8	0.0347	0.0589	0.0936	803.0	241.2	1044.2	0.9755	0.2098	1.1852	690
700°	3096.4	0.0394	0.0353	0.0747	846.3	157.0	1003.2	1.0117	0.1354	1.1471	700°
706.1	3226.0	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	706.1

(Courtesy John Wiley & Sons, Inc.) Abridged from Keenan, "Steam Tables and (A.S.M.E.)

TABLE 2.—SATURATED STEAM: PRESSURE TABLE[1]
(English units)

Abs press. lb per sq in.	Temp, °F	Specific volume			Total heat			Entropy			Abs press. lb per sq in.
		Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
0.0887	32	0.01602	3,301	3,301	0.00	1,073.4	1,073.4	0.0000	2.1834	2.1834	**0.0887**
0.1217	40	0.01602	2,445	2,445	8.05	1,069.1	1,077.1	0.0162	2.1397	2.1559	**0.1217**
0.1780	50	0.01602	1,704.8	1,704.8	18.06	1,063.6	1,081.7	0.0361	2.0870	2.1231	**0.1780**
½ in. Hg	58.83	0.01603	1,256.9	1,256.9	26.88	1,058.8	1,085.7	0.0533	2.0422	2.0955	**½ in. Hg**
0.2561	60	0.01603	1,208.0	1,208.0	28.05	1,058.2	1,086.2	0.0555	2.0364	2.0919	**0.2561**
¾ in. Hg	70.44	0.01605	856.5	856.5	38.47	1,052.5	1,091.0	0.0754	1.9856	2.0609	**¾ in. Hg**
1 in. Hg	79.06	0.01607	652.7	652.7	47.06	1,047.8	1,094.9	0.0914	1.9451	2.0365	**1 in. Hg**
1½ in. Hg	91.75	0.01610	445.3	445.3	59.72	1,040.8	1,100.6	0.1147	1.8877	2.0024	**1½ in. Hg**
2 in. Hg	101.17	0.01613	339.5	339.5	69.10	1,035.7	1,104.8	0.1316	1.8468	1.9784	**2 in. Hg**
1.0	101.76	0.01614	333.8	333.9	69.69	1,035.3	1,105.0	0.1326	1.8442	1.9769	**1.0**
2½ in. Hg	108.73	0.01616	275.2	275.2	76.63	1,031.5	1,108.1	0.1450	1.8148	1.9598	**2½ in. Hg**
3 in. Hg	115.08	0.01618	231.8	231.8	82.96	1,027.9	1,110.8	0.1561	1.7885	1.9446	**3 in. Hg**
2.0	126.10	0.01623	173.94	173.96	93.97	1,021.6	1,115.6	0.1750	1.7442	1.9192	**2.0**
3.0	141.49	0.01630	118.84	118.86	109.33	1,012.7	1,122.0	0.2009	1.6847	1.8856	**3.0**
4.0	152.99	0.01636	90.72	90.74	120.83	1,005.9	1,126.8	0.2198	1.6420	1.8618	**4.0**
5.0	162.25	0.01641	73.59	73.61	130.10	1,000.4	1,130.6	0.2348	1.6088	1.8435	**5.0**
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1,133.7	0.2473	1.5814	1.8287	**6.0**
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1,136.4	0.2580	1.5582	1.8162	**7.0**
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1,138.9	0.2674	1.5379	1.8053	**8.0**
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1,141.0	0.2758	1.5200	1.7958	**9.0**
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1,143.0	0.2834	1.5040	1.7874	**10.0**
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1,144.8	0.2903	1.4894	1.7797	**11.0**
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1,146.4	0.2968	1.4760	1.7727	**12.0**
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1,147.9	0.3027	1.4636	1.7663	**13.0**
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1,149.3	0.3082	1.4521	1.7604	**14.0**
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1,150.2	0.3119	1.4446	1.7564	**14.696**

15.0	213.03	0.01671	26.29	26.31	181.04	969.6	1,150.6	0.3134	1.4414	1.7548	**15.0**
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1,151.8	0.3184	1.4312	1.7496	**16.0**
17.0	219.43	0.01676	23.38	23.40	187.48	965.4	1,152.9	0.3230	1.4218	1.7448	**17.0**
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1,154.0	0.3274	1.4127	1.7402	**18.0**
19.0	225.23	0.01680	21.07	21.08	193.34	961.7	1,155.0	0.3316	1.4042	1.7358	**19.0**
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1,156.0	0.3356	1.3960	1.7317	**20.0**
21.0	230.56	0.01684	19.180	19.197	198.72	958.2	1,156.9	0.3395	1.3883	1.7278	**21.0**
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1,157.8	0.3431	1.3809	1.7240	**22.0**
23.0	235.49	0.01687	17.613	17.630	203.70	955.0	1,158.6	0.3466	1.3738	1.7204	**23.0**
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1,159.5	0.3500	1.3670	1.7170	**24.0**
25.0	240.07	0.01690	16.289	16.306	208.33	951.9	1,160.2	0.3533	1.3604	1.7137	**25.0**
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1,161.0	0.3564	1.3542	1.7106	**26.0**
27.0	244.36	0.01694	15.155	15.172	212.67	949.0	1,161.7	0.3594	1.3481	1.7075	**27.0**
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1,162.4	0.3624	1.3422	1.7046	**28.0**
29.0	248.40	0.01697	14.173	14.190	216.77	946.3	1,163.1	0.3652	1.3365	1.7018	**29.0**
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1,163.7	0.3680	1.3310	1.6990	**30.0**
31.0	252.22	0.01700	13.313	13.330	220.64	943.7	1,164.4	0.3707	1.3257	1.6964	**31.0**
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1,165.0	0.3732	1.3206	1.6938	**32.0**
33.0	255.84	0.01703	12.556	12.573	224.32	941.2	1,165.6	0.3758	1.3156	1.6914	**33.0**
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1,166.1	0.3783	1.3107	1.6890	**34.0**
35.0	259.28	0.01706	11.881	11.898	227.82	938.9	1,166.7	0.3807	1.3060	1.6866	**35.0**
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1,167.2	0.3830	1.3014	1.6844	**36.0**
37.0	262.57	0.01708	11.277	11.294	231.17	936.6	1,167.8	0.3853	1.2968	1.6822	**37.0**
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1,168.3	0.3876	1.2925	1.6800	**38.0**
39.0	265.72	0.01711	10.733	10.750	234.38	934.4	1,168.8	0.3898	1.2882	1.6780	**39.0**
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1,169.2	0.3919	1.2840	1.6759	**40.0**
41.0	268.74	0.01713	10.240	10.257	237.45	932.3	1,169.7	0.3940	1.2799	1.6739	**41.0**
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1,170.2	0.3961	1.2759	1.6720	**42.0**
43.0	271.65	0.01716	9.791	9.808	240.42	930.2	1,170.6	0.3981	1.2720	1.6701	**43.0**
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1,171.1	0.4000	1.2682	1.6683	**44.0**
45.0	274.45	0.01718	9.382	9.399	243.28	928.2	1,171.5	0.4020	1.2645	1.6665	**45.0**
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1,171.9	0.4039	1.2608	1.6647	**46.0**
47.0	277.14	0.01720	9.006	9.023	246.03	926.3	1,172.3	0.4057	1.2572	1.6630	**47.0**
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1,172.7	0.4076	1.2537	1.6613	**48.0**
49.0	279.74	0.01723	8.659	8.676	248.68	924.4	1,173.1	0.4093	1.2503	1.6596	**49.0**
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1,173.5	0.4111	1.2469	1.6580	**50.0**
51.0	282.26	0.01725	8.340	8.357	251.26	922.6	1,173.9	0.4128	1.2436	1.6564	**51.0**
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1,174.3	0.4145	1.2404	1.6549	**52.0**
53.0	284.70	0.01727	8.043	8.060	253.76	920.9	1,174.6	0.4162	1.2372	1.6533	**53.0**
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1,175.0	0.4178	1.2340	1.6518	**54.0**

[1] ALLEN and BURSLEY, "Heat Engines."

TABLE 2.—SATURATED STEAM: PRESSURE TABLE.[1]—(*Continued*)

Abs press. lb per sq in.	Temp, °F	Specific volume			Total heat			Entropy			Abs press. lb per sq in.
		Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
55.0	287.07	0.01729	7.766	7.783	256.19	919.1	1,175.3	0.4194	1.2309	1.6504	**55.0**
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1,175.7	0.4210	1.2279	1.6489	**56.0**
57.0	289.37	0.01732	7.509	7.527	258.55	917.5	1,176.0	0.4226	1.2249	1.6475	**57.0**
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1,176.4	0.4241	1.2220	1.6461	**58.0**
59.0	291.62	0.01734	7.270	7.287	260.86	915.8	1,176.7	0.4256	1.2191	1.6447	**59.0**
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1,177.0	0.4271	1.2162	1.6434	**60.0**
61.0	293.79	0.01736	7.044	7.062	263.09	914.2	1,177.3	0.4286	1.2134	1.6420	**61.0**
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1,177.6	0.4300	1.2107	1.6407	**62.0**
63.0	295.91	0.01738	6.833	6.850	265.27	912.7	1,177.9	0.4315	1.2080	1.6394	**63.0**
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1,178.2	0.4329	1.2053	1.6382	**64.0**
65.0	297.97	0.01740	6.634	6.652	267.39	911.1	1,178.5	0.4343	1.2026	1.6369	**65.0**
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1,178.8	0.4356	1.2001	1.6357	**66.0**
67.0	299.99	0.01742	6.447	6.464	269.47	909.6	1,179.1	0.4370	1.1975	1.6345	**67.0**
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1,179.4	0.4384	1.1950	1.6333	**68.0**
69.0	301.96	0.01744	6.271	6.288	271.50	908.2	1,179.6	0.4397	1.1924	1.6321	**69.0**
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1,179.9	0.4410	1.1900	1.6310	**70.0**
71.0	303.88	0.01746	6.104	6.121	273 48	906.7	1,180.2	0.4423	1.1876	1.6298	**71.0**
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1,180.5	0.4435	1.1852	1.6287	**72.0**
73.0	305.76	0.01747	5.946	5.963	275.42	905.3	1,180.7	0.4448	1.1828	1.6276	**73.0**
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1,181.0	0.4460	1.1805	1.6265	**74.0**
75.0	307.60	0.01749	5.796	5.813	277.32	903.9	1,181.2	0.4473	1.1782	1.6254	**75.0**
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1,181.5	0.4485	1.1759	1.6244	**76.0**
77.0	309.39	0.01751	5.653	5.671	279.17	902.6	1,181.7	0.4497	1.1736	1.6233	**77.0**
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1,182.0	0.4509	1.1714	1.6223	**78.0**
79.0	311.16	0.01753	5.517	5.535	281.00	901.2	1,182.2	0.4520	1.1692	1.6212	**79.0**
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1,182.4	0.4532	1.1670	1.6202	**80.0**
81.0	312.88	0.01755	5.388	5.406	282.79	899.9	1,182.7	0.4544	1.1649	1.6192	**81.0**
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1,182.9	0.4555	1.1627	1.6182	**82.0**
83.0	314.58	0.01756	5.264	5.282	284.56	898.6	1,183.1	0.4566	1.1606	1.6173	**83.0**
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1,183.4	0.4578	1.1586	1.6163	**84.0**

85.0	316.25	0.01758	5.146	5.164	286.28	897.3	1,183.6	0.4589	1.1565	1.6154	**85.0**
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1,183.8	0.4599	1.1545	1.6144	**86.0**
87.0	317.88	0.01760	5.033	5.051	287.97	896.0	1,184.0	0.4610	1.1524	1.6135	**87.0**
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1,184.2	0.4621	1.1505	1.6126	**88.0**
89.0	319.48	0.01762	4.926	4.944	289.63	894.8	1,184.4	0.4632	1.1485	1.6116	**89.0**
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1,184.6	0.4642	1.1465	1.6107	**90.0**
91.0	321.05	0.01764	4.823	4.841	291.26	893.6	1,184.8	0.4652	1.1446	1.6099	**91.0**
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1,185.0	0.4663	1.1427	1.6090	**92.0**
93.0	322.60	0.01765	4.724	4.742	292.87	892.4	1,185.2	0.4673	1.1408	1.6081	**93.0**
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1,185.4	0.4683	1.1389	1.6072	**94.0**
95.0	324.13	0.01767	4.629	4.647	294.47	891.2	1,185.6	0.4693	1.1370	1.6064	**95.0**
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1,185.8	0.4703	1.1352	1.6055	**96.0**
97.0	325.62	0.01768	4.539	4.557	296.02	890.0	1,186.0	0.4713	1.1334	1.6047	**97.0**
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1,186.2	0.4723	1.1316	1.6038	**98.0**
99.0	327.10	0.01770	4.450	4.468	297.57	888.8	1,186.4	0.4732	1.1298	1.6030	**99.0**
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1,186.6	0.4742	1.1280	1.6022	**100.0**
101.0	328.55	0.01772	4.367	4.385	299.08	887.7	1,186.8	0.4751	1.1262	1.6014	**101.0**
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1,186.9	0.4761	1.1245	1.6006	**102.0**
103.0	329.98	0.01773	4.287	4.305	300.57	886.6	1,187.1	0.4770	1.1228	1.5998	**103.0**
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1,187.3	0.4779	1.1211	1.5990	**104.0**
105.0	331.38	0.01775	4.208	4.226	302.03	885.4	1,187.5	0.4789	1.1194	1.5982	**105.0**
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1,187.6	0.4798	1.1177	1.5974	**106.0**
107.0	332.76	0.01777	4.134	4.152	303.48	884.3	1,187.8	0.4807	1.1160	1.5967	**107.0**
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1,188.0	0.4816	1.1144	1.5959	**108.0**
109.0	334.12	0.01778	4.061	4.079	304.90	883.2	1,188.1	0.4825	1.1127	1.5952	**109.0**
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1,188.3	0.4834	1.1111	1.5944	**110.0**
111.0	335.46	0.01780	3.992	4.010	306.31	882.1	1,188.4	0.4842	1.1095	1.5937	**111.0**
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1,188.6	0.4851	1.1079	1.5930	**112.0**
113.0	336.78	0.01781	3.925	3.943	307.68	881.1	1,188.8	0.4859	1.1063	1.5922	[illegible]
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1,188.9	0.4868	1.1048	1.5915	
115.0	338.08	0.01783	3.860	3.878	309.04	880.0	1,189.1	0.4876	1.1032	1.5908	[illegible]
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1,189.2	0.4885	1.1017	1.5901	**116.0**
117.0	339.37	0.01784	3.797	3.815	310.38	879.0	1,189.4	0.4893	1.1001	1.5894	**117.0**
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1,189.5	0.4901	1.0986	1.5887	[illegible]
119.0	340.64	0.01786	3.736	3.754	311.71	878.0	1,189.7	0.4909	1.0971	1.5880	
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1,189.8	0.4918	1.0956	1.5874	
121.0	341.88	0.01787	3.679	3.697	313.02	876.9	1,190.0	0.4926	1.0941	1.5867	
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1,190.1	0.4934	1.0926	1.5860	
123.0	343.12	0.01789	3.624	3.642	314.32	875.9	1,190.2	0.4942	1.0911	1.5853	
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1,190.4	0.4950	1.0897	1.5847	

[1] Allen and Bursley, "Heat Engines."

TABLE 2.—SATURATED STEAM: PRESSURE TABLE.[1]—(*Continued*)

Abs press. lb per sq in.	Temp, °F	Specific volume			Total heat			Entropy			Abs press. lb per sq in.
		Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
125.0	344.34	0.01790	3.569	3.587	315.60	874.9	1,190.5	0.4958	1.0882	1.5840	**125.0**
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1,190.6	0.4965	1.0868	1.5834	**126.0**
127.0	345.54	0.01792	3.514	3.532	316.86	873.9	1,190.8	0.4973	1.0854	1.5827	**127.0**
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1,190.9	0.4981	1.0840	1.5821	**128.0**
129.0	346.73	0.01793	3.459	3.477	318.11	872.9	1,191.0	0.4989	1.0826	1.5814	**129.0**
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1,191.2	0.4996	1.0812	1.5808	**130.0**
131.0	347.90	0.01794	3.408	3.426	319.34	872.0	1,191.3	0.5004	1.0798	1.5802	**131.0**
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1,191.4	0.5011	1.0784	1.5796	**132.0**
133.0	349.06	0.01796	3.359	3.377	320.56	871.0	1,191.6	0.5019	1.0770	1.5789	**133.0**
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1,191.7	0.5026	1.0757	1.5783	**134.0**
135.0	350.21	0.01797	3.311	3.329	321.77	870.0	1,191.8	0.5034	1.0743	1.5777	**135.0**
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1,191.9	0.5041	1.0730	1.5771	**136.0**
137.0	351.35	0.01799	3.265	3.283	322.97	869.1	1,192.0	0.5048	1.0716	1.5765	**137.0**
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1,192.2	0.5056	1.0703	1.5759	**138.0**
139.0	352.47	0.01800	3.220	3.238	324.15	868.1	1,192.3	0.5063	1.0690	1.5753	**139.0**
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1,192.5	0.5070	1.0677	1.5747	**140.0**
141.0	353.59	0.01802	3.176	3.194	325.33	867.2	1,192.5	0.5077	1.0664	1.5741	**141.0**
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1,192.6	0.5084	1.0651	1.5735	**142.0**
143.0	354.68	0.01803	3.133	3.151	326.49	866.3	1,192.8	0.5091	1.0638	1.5729	**143.0**
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1,192.9	0.5098	1.0625	1.5724	**144.0**
145.0	355.77	0.01804	3.092	3.110	327.63	865.3	1,193.0	0.5105	1.0612	1.5718	**145.0**
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1,193.1	0.5112	1.0600	1.5712	**146.0**
147.0	356.84	0.01806	3.051	3.069	328.76	864.4	1,193.2	0.5119	1.0587	1.5707	**147.0**
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1,193.3	0.5126	1.0575	1.5701	**148.0**
149.0	357.90	0.01807	3.012	3.030	329.88	863.5	1,193.4	0.5133	1.0563	1.5695	**149.0**
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1,193.5	0.5140	1.0550	1.5690	**150.0**
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1,193.7	0.5153	1.0526	1.5679	**152.0**
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1,193.9	0.5166	1.0502	1.5668	**154.0**
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1,194.1	0.5180	1.0478	1.5658	**156.0**
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1,194.3	0.5193	1.0454	1.5647	**158.0**

160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1,194.5	0.5205	1.0431	1.5636	**160.0**
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1,194.7	0.5218	1.0408	1.5626	**162.0**
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1,194.9	0.5230	1.0385	1.5616	**164.0**
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1,195.1	0.5243	1.0363	1.5606	**166.0**
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1,195.3	0.5255	1.0340	1.5596	**168.0**
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1,195.4	0.5268	1.0318	1.5586	**170.0**
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1,195.6	0.5280	1.0296	1.5576	**172.0**
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1,195.8	0.5292	1.0275	1.5566	**174.0**
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1,196.0	0.5304	1.0253	1.5557	**176.0**
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1,196.1	0.5315	1.0232	1.5548	**178.0**
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1,196.3	0.5327	1.0211	1.5538	**180.0**
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1,196.4	0.5339	1.0190	1.5529	**182.0**
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1,196.6	0.5350	1.0169	1.5520	**184.0**
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1,196.8	0,5362	1.0149	1.5511	**186.0**
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1,196.9	0.5373	1.0129	1.5502	**188.0**
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1,197.0	0.5384	1.0109	1.5493	**190.0**
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1,197.2	0.5395	1.0089	1.5484	**192.0**
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1,197.3	0.5406	1.0070	1.5475	**194.0**
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1,197.5	0.5417	1.0050	1.5467	**196.0**
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1,197.6	0.5427	1.0031	1.5458	**198.0**
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1,197.8	0.5438	1.0012	1.5450	**200.0**
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1,198.1	0.5465	0.9964	1.5429	**205.0**
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1,198.4	0.5491	0.9918	1.5409	**210.0**
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1,198.7	0.5516	0.9873	1.5389	**215.0**
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1,199.0	0.5540	0.9829	1.5369	**220.0**
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1,199.3	0.5565	0.9786	1.5350	**225.0**
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1,199.6	0.5588	0.9743	1.5332	**230.0**
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1,199.8	0.5612	0.9702	1.5313	**235.0**
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1,200.1	0.5635	0.9661	1.5295	**240.0**
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1,200.3	0.5658	0.9620	1.5278	**245.0**
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1,200.5	0.5680	0.9581	1.5261	**250.0**
255.0	402.71	0.01869	1.7873	1.8060	377.91	822.9	1,200.8	0.5701	0.9542	1.5244	**255.0**
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1,201.0	0.5723	0.9504	1.5227	**260.0**
265.0	406.12	0.01874	1.7210	1.7397	381.62	819.6	1,201.2	0.5744	0.9467	1.5210	**265.0**
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1,201.4	0.5765	0.9430	1.5194	**270.0**
275.0	409.44	0.01880	1.6593	1.6781	385.24	816.3	1,201.6	0.5785	0.9393	1.5178	**275.0**
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1,201.8	0.5805	0.9357	1.5163	**280.0**
285.0	412.66	0.01884	1.6018	1.6206	388.77	813.2	1,201.9	0.5825	0.9322	1.5147	**285.0**
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1,202.1	0.5845	0.9287	1.5132	**290.0**
295.0	415.80	0.01889	1.5480	1.5669	392.21	810.0	1,202.2	0.5864	0.9253	1.5117	**295.0**

[1] ALLEN and BURSLEY, "Heat Engines."

TABLE 2.—SATURATED STEAM: PRESSURE TABLE.[1]—(*Continued*)

Abs press. lb per sq in.	Temp, °F	Specific volume			Total heat			Entropy			Abs press. lb per sq in.
		Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1,202.4	0.5883	0.9220	1.5102	**300.0**
310.0	420.35	0.01896	1.4738	1.4928	397.23	805.5	1,202.7	0.5920	0.9153	1.5074	**310.0**
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1,203.0	0.5957	0.9089	1.5046	**320.0**
330.0	426.16	0.01905	1.3849	1.4039	403.64	799.5	1,203.2	0.5992	0.9026	1.5018	**330.0**
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1,203.4	0.6027	0.8965	1.4992	**340.0**
350.0	431.71	0.01914	1.3054	1.3245	409.81	793.7	1,203.6	0.6061	0.8905	1.4966	**350.0**
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1,203.7	0.6094	0.8846	1.4940	**360.0**
370.0	437.01	0.01923	1.2344	1.2536	415.73	788.1	1,203.8	0.6126	0.8789	1.4915	**370.0**
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1,203.9	0.6157	0.8733	1.4891	**380.0**
390.0	442.11	0.01931	1.1705	1.1898	421.44	782.6	1,204.0	0.6188	0.8678	1.4867	**390.0**
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1,204.1	0.6218	0.8625	1.4843	**400.0**
410.0	447.00	0.0194	1.1123	1.1317	426.9	777.2	1,204.1	0.6248	0.8572	1.4820	**410.0**
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1,204.1	0.6277	0.8520	1.4798	**420.0**
430.0	451.72	0.0195	1.0593	1.0788	432.3	771.9	1,204.1	0.6306	0.8470	1.4775	**430.0**
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1,204.1	0.6334	0.8420	1.4753	**440.0**
450.0	456.27	0.0196	1.0107	1.0303	437.4	766.7	1,204.1	0.6361	0.8371	1.4732	**450.0**
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1,204.0	0.6388	0.8322	1.4711	**460.0**
470.0	460.66	0.0196	0.9665	0.9861	442.4	761.6	1,204.0	0.6415	0.8275	1.4690	**470.0**
480.0	462.80	0.0197	0.9456	0.9653	444.9	759.0	1,203.9	0.6441	0.8228	1.4670	**480.0**
490.0	464.91	0.0197	0.9256	0.9453	447.3	756.5	1,203.8	0.6467	0.8182	1.4650	**490.0**
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1,203.7	0.6493	0.8137	1.4630	**500.0**
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1,203.5	0.6543	0.8048	1.4591	**520.0**
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1,203.2	0.6592	0.7962	1.4554	**540.0**
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1,202.9	0.6639	0.7878	1.4517	**560.0**
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1,202.5	0.6686	0.7796	1.4482	**580.0**
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1,202.1	0.6731	0.7716	1.4447	**600.0**
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1,201.7	0.6775	0.7638	1.4413	**620.0**
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1,201.2	0.6818	0.7562	1.4380	**640.0**
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1,200.8	0.6861	0.7487	1.4348	**660.0**
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1,200.2	0.6902	0.7414	1.4316	**680.0**

700.0	503.04	0 0206	0 6321	0 6527	492.9	706 8	1,199 7	0 6943	0 7342	1 4285	700.0
720 0	506 19	0 0206	0 6128	0 6334	496 8	702 4	1,199 2	0 6983	0 7272	1 4255	720.0
740 0	509 28	0 0207	0 5944	0 6151	500 6	697 9	1,198 6	0 7022	0 7203	1 4225	740.0
760 0	512 30	0 0208	0 5769	0 5977	504 4	693 5	1,198 0	0 7060	0 7136	1 4196	760 0
780 0	515 27	0 0209	0 5602	0 5811	508 2	689 2	1 197 4	0 7098	0 7069	1 4167	780.0
800 0	518 18	0 0209	0 5444	0 5653	511 8	684 9	1,196 7	0 7135	0 7004	1 4139	800 0
820 0	521 03	0 0210	0 5293	0 5503	515 5	680 6	1,196 0	0 7171	0 6940	1 4111	820 0
840 0	523 83	0 0211	0 5149	0 5360	519 0	676 4	1,195 4	0 7207	0 6877	1 4084	840 0
860 0	526 58	0 0212	0 5013	0 5225	522 6	672 1	1 194 7	0 7242	0 6815	1 4057	860 0
880.0	529 29	0 0213	0 4881	0 5094	526 0	667 9	1,194 0	0 7277	0 6754	1 4031	880.0
900 0	531 95	0 0213	0 4756	0 4969	529 5	663 8	1,193 3	0.7311	0 6694	1 4005	900 0
920 0	534 56	0 0214	0 4635	0 4849	532 9	659 7	1,192 6	0 7344	0 6635	1 3980	920.0
940 0	537 13	0 0215	0 4520	0 4735	536 2	655 6	1,191 8	0 7377	0 6577	1 3954	940 0
960 0	539 66	0 0216	0 4409	0 4625	539 6	651 5	1,191 1	0 7410	0 6520	1 3930	960.0
980 0	542 14	0 0217	0 4303	0 4520	542 8	647 5	1,190 3	0 7442	0 6464	1 3905	980.0
1,000 0	544 58	0 0217	0 4202	0 4419	546 0	643 5	1,189 6	0 7473	0 6408	1 3881	1,000.0
1,050 0	550 53	0 0219	0 3960	0 4179	554 0	633 6	1,187 6	0 7550	0 6273	1 3822	1,050.0
1,100 0	556 28	0 0222	0 3738	0 3960	561 7	623 9	1,185 6	0 7624	0 6141	1 3765	1,100.0
1,150 0	561 81	0 0224	0 3540	0 3764	569 2	614 3	1 183 5	0 7695	0 6014	1 3709	1,150 0
1,200.0	567.14	0 0226	0 3356	0 3582	576 5	604 9	1,181 4	0 7764	0 5891	1 3656	1,200.0
1,250 0	572 30	0 0228	0 3187	0 3415	583 6	595 6	1,179 2	0 7831	0 5772	1.3603	1,250 0
1,300 0	577 32	0 0230	0 3029	0 3259	590 6	586 3	1 177 0	0 7897	0 5654	1.3552	1,300 0
1,350.0	582 21	0 0232	0 2884	0 3116	597 5	577 2	1,174 7	0 7962	0 5540	1 3501	1,350.0
1,400 0	586 96	0 0235	0 2748	0 2983	604 3	568 1	1,172 4	0 8024	0 5428	1 3452	1,400 0
1,450 0	591 58	0 0237	0 2621	0 2858	611 0	559 1	1,170 0	0 8086	0 5318	1.3404	1,450 0
1,500 0	596 08	0 0239	0 2502	0 2741	617 5	550 2	1,167 6	0 8146	0 5212	1.3357	1,500 0
1,600 0	604 74	0 0244	0 2284	0 2528	630 2	532 6	1,162 7	0 8262	0 5003	1 3265	1,600 0
1,700.0	612 98	0 0249	0 2089	0 2338	642 5	515 0	1,157 5	0 8373	0 4801	1 3174	1,700.0
1,800.0	620 86	0 0254	0 1913	0 2167	654 7	497 2	1,151 8	0 8482	0 4601	1 3083	1,800 0
1,900.0	628 39	0 0260	0 1754	0 2014	666 8	478 9	1,145 7	0 8589	0 4402	1 2990	1,900.0
2,000 0	635 6	0 0265	0.1610	0 1875	679 0	460 0	1,139 0	0 8696	0 4200	1 2896	2,000 0
2,100 0	642 6	0 0271	0.1473	0 1744	691 3	440 4	1,131 7	0 8804	0 3996	1.2800	2,100.0
2,200 0	649 2	0 0277	0.1346	0 1623	703 7	420 0	1,123 8	0 8912	0 3788	1 2700	2,200.0
2,300 0	655 7	0 0284	0.1226	0 1510	716 4	398 7	1,115 2	0 9021	0 3575	1 2596	2,300.0
2,400 0	661 9	0 0292	0 1112	0 1404	729 4	376 4	1,105 8	0 9133	0.3356	1 2488	2,400.0
2,500 0	668 0	0 0301	0 1002	0 1303	742 8	352 8	1,095 6	0 9247	0 3129	1 2375	2,500 0
2,600 0	673 8	0 0310	0 0895	0 1205	756 7	327 8	1,084 5	0 9364	0 2892	1 2257	2,600.0
2,700 0	679 5	0 0321	0 0790	0 1111	771 2	301 2	1,072 4	0 9487	0 2644	1 2131	2,700.0
2,800 0	684 9	0 0333	0 0688	0 1021	786 7	272 3	1,058 9	0 9618	0 2379	1 1996	2,800.0
2,900 0	690 2	0 0348	0 0585	0 0933	803 6	240 0	1,043 7	0 9760	0 2088	1 1847	2,900.0
3,000 0	695 2	0 0367	0 0477	0 0844	823 1	202 5	1,025 6	0 9922	0 1754	1 1676	3,000.0
3,100 0	700 2	0 0395	0 0348	0 0743	847 2	155 0	1,002 2	1 0126	0 1336	1 1462	3,100.0
3,200 0	704 9	0 0459	0 0142	0 0601	887 0	75 9	962 9	1 0461	0 0651	1 1112	3,200.0
3,226.0	706.1	0 0522	0	0 0522	925 0	0	925 0	1 0785	0	1 0785	3,226.0

[1] Allen and Bursley, "Heat Engines"

TABLE 3.—SUPER
(Engli

Abs press. lb per sq in. (Sat. temp.)		Sat. water	Sat. steam	Temperature— 220°	240°	260°	280°	300°	320°	340°	360°	380°
1	v	0.0	333.9	404.4	416.4	428.2	440.2	452.1	464.0	475.9	487.8	499.7
	h	69.7	1,105.0	1,158.8	1,167.8	1,176.9	1,185.9	1,195.0	1,204.1	1,213.2	1,222.4	1,231.5
(101.76)	s	0.1326	1.9769	2.0638	2.0770	2.0898	2.1021	2.1142	2.1260	2.1376	2.1489	2.1599
14.696	v	0.02	26.82	27.16	27.98	28.82	29.67	30.52	31.34	32.17	33.00	33.82
	h	180.0	1,150.2	1,154.1	1,163.8	1,173.4	1,182.8	1,192.2	1,201.6	1,210.9	1,220.3	1,229.6
(212.00)	s	0.3119	1.7564	1.7623	1.7764	1.7898	1.8028	1.8154	1.8275	1.8394	1.8509	1.8621
25	v	0.017	16.306			16.821	17.331	17.838	18.337	18.833	19.324	19.812
	h	208.3	1,160.2			1,170.3	1,180.2	1,190.0	1,199.6	1,209.2	1,218.7	1,228.2
(240.07)	s	0.3533	1.7137			1.7280	1.7416	1.7544	1.7671	1.7792	1.7909	1.8023
40	v	0.017	10.497				10.714	11.044	11.367	11.686	11.999	12.312
	h	235.9	1,169.2				1,176.1	1,186.5	1,196.5	1,206.4	1,216.2	1,225.9
(267.24)	s	0.3919	1.6759				1.6853	1.6990	1.7122	1.7248	1.7369	1.7486
50	v	0.017	8.514					8.777	9.042	9.303	9.558	9.810
	h	250.0	1,173.5					1,183.9	1,194.4	1,204.5	1,214.5	1,224.4
(281.01)	s	0.4111	1.6580					1.6718	1.6854	1.6983	1.7107	1.7226
65	v	0.017	6.652					6.674	6.890	7.099	7.302	7.500
	h	267.4	1,178.5					1,179.7	1,190.9	1,201.6	1,211.9	1,222.1
(297.97)	s	0.4343	1.6369					1.6385	1.6530	1.6666	1.6793	1.6916
75	v	0.017	5.813						5.932	6.119	6.298	6.472
	h	277.3	1,181.2						1,188.4	1,199.5	1,210.1	1,220.5
(307.60)	s	0.4473	1.6254						1.6346	1.6487	1.6618	1.6743
90	v	0.018	4.892							5.055	5.208	5.357
	h	290.4	1,184.6							1,196.2	1,207.2	1,218.0
(320.27)	s	0.4642	1.6107							1.6253	1.6390	1.6519
100	v	0.018	4.426							4.520	4.663	4.800
	h	298.3	1,186.6							1,193.9	1,205.3	1,216.2
(327.83)	s	0.4742	1.6022							1.6114	1.6254	1.6387
114	v	0.018	3.910							3.928	4.059	4.182
	h	308.4	1,188.9							1,190.5	1,202.4	1,213.7
(337.43)	s	0.4868	1.5915							1.5936	1.6082	1.6219
126	v	0.018	3.560								3.647	3.762
	h	316.2	1,190.6								1,199.9	1,211.6
(344.94)	s	0.4965	1.5834								1.5947	1.6088
140	v	0.018	3.216								3.255	11.295
	h	324.7	1,192.4								1,196.8	1,220.6
(353.03)	s	0.5070	1.5747								1.5801	1.7344
150	v	0.018	3.010									3.122
	h	330.4	1,193.5									1,207.0
(358.43)	s	0.5140	1.5690									1.5852
164	v	0.018	2.764									2.834
	h	338.0	1,194.9									1,204.2
(365.52)	s	0.5230	1.5616									1.5727
176	v	0.018	2.584									2.624
	h	344.0	1,196.0									1,201.7
(371.24)	s	0.5304	1.5557									1.5626
190	v	0.018	2.401									2.411
	h	350.8	1,197.0									1,198.7
(377.55)	s	0.5384	1.5493									1.5513
200	v	0.018	2.285									
	h	355.3	1,197.8									
(381.82)	s	0.5438	1.5450									
214	v	0.018	2.141									
	h	361.5	1,198.6									
(387.53)	s	0.5511	1.5393									

[1] ALLEN and BURSLEY,

HEATED STEAM[1]
sh units)

degrees Fahrenheit												
400°	420°	440°	460°	480°	500°	600°	700°	800°	900°	1000°		Abs press. lb-sq in. (Sat. temp.)
511.7	523.6	535.6	547.5	559.4	571.3	630.9	690.6	750.2	809.8	869.4	v	
1,240.7	1,250.0	1,259.2	1,268.5	1,277.9	1,287.2	1,334.6	1,383.0	1,432.6	1,483.3	1,535.2	h	1
2.1707	2.1813	2.1917	2.2019	2.2120	2.2218	2.2688	2.3125	2.3535	2.3922	2.4291	s	(101.76)
34.65	35.47	36.29	37.11	37.93	38.75	42.83	46.91	50.97	55.03	59.09	v	
1,239.0	1,248.4	1,257.7	1,267.1	1,276.6	1,286.0	1,333.7	1,382.4	1,432.1	1,482.9	1,534.9	h	14.696
1.8731	1.8839	1.8944	1.9048	1.9149	1.9249	1.9722	2.0161	2.0572	2.0961	2.1330	s	(212.00)
20.30	20.79	21.27	21.76	22.25	22.73	25.15	27.55	29.94	32.33	34.73	v	
1,237 6	1,247.1	1,256.6	1,266.1	1,275.6	1,285.1	1,333.0	1,381.9	1,431.8	1,482.7	1,534.7	h	25
1.8134	1.8243	1.8350	1.8454	1.8556	1.8657	1.9132	1.9572	1.9985	2.0374	2.0743	s	(240.07)
12.623	12.932	13.238	13.547	13.855	14.161	15.682	17.190	18.686	20.18	21.68	v	
1,235.6	1,245.3	1,254.9	1,264.5	1,274.1	1,283.7	1,332.0	1,381.2	1,431.3	1,482.3	1,534.4	h	40
1.7599	1.7710	1.7818	1.7924	1.8027	1.8128	1.8607	1.9050	1.9464	1.9854	2.0224	s	(267.24)
10.061	10.313	10.560	10.809	11.058	11.304	12.527	13.737	14.934	16.136	17.337	v	
1,234.2	1,244.0	1,253.7	1,263.4	1,273.1	1,282.8	1,331.4	1,380.7	1,431.0	1,482.1	1,534.1	h	50
1.7341	1.7454	1.7563	1.7669	1.7773	1.7875	1.8357	1.8801	1.9217	1.9608	1.9977	s	(281.01)
7.696	7.892	8.085	8.279	8.473	8.665	9.614	10.550	11.475	12.401	13.325	v	
1,232 1	1,242.1	1,252.0	1,261.8	1,271.6	1,281.4	1,330.3	1,380.0	1,430.5	1,481.7	1,533.8	h	65
1.7033	1.7148	1.7259	1.7367	1.7472	1.7576	1.8060	1.8508	1.8926	1.9317	1.9687	s	(297.97)
6.644	6.817	6.985	7.155	7.324	7.492	8.319	9.133	9.938	10.741	11.542	v	
1,230 7	1,240.8	1,250.8	1,260.7	1,270.6	1,280.4	1,329.7	1,379.5	1,430.1	1,481.4	1,533.6	h	75
1.6863	1.6979	1.7091	1.7200	1.7307	1.7410	1.7898	1.8347	1.8766	1.9158	1.9528	s	(307.60)
5.504	5.651	5.794	5.938	6.080	6.220	6.916	7.599	8.272	8.943	9.610	v	
1,228.5	1,238.8	1,248.9	1,259.0	1,269.0	1,279.0	1,328.6	1,378.7	1,429.6	1,481.0	1,533.3	h	90
1.6642	1.6760	1.6875	1.6986	1.7094	1.7198	1.7690	1.8142	1.8563	1.8955	1.9326	s	(320.27)
4 934	5.068	5.198	5.328	5.457	5.585	6.215	6.831	7.439	8.044	8.644	v	
1,226 9	1,237.4	1,247.7	1,257.9	1,268.0	1,278.0	1,327.9	1,378.2	1,429.2	1,480.7	1,533.1	h	100
1.6512	1.6633	1.6748	1.6860	1.6969	1.7075	1.7569	1.8023	1.8445	1.8838	1.9209	s	(327.83)
4.303	4.423	4.539	4.654	4.769	4.883	5.439	5.983	6.518	7.048	7.577	v	
1,224 7	1,235.4	1,245.9	1,256.3	1,266.5	1,276.7	1,326.9	1,377.5	1,428.7	1,480.4	1,532.8	h	114
1.6348	1.6471	1.6589	1.6702	1.6813	1.6920	1.7418	1.7874	1.8297	1.8692	1.9064	s	(337.43)
3.873	3.984	4.090	4.196	4.301	4.405	4.911	5.406	5.892	6.372	6.851	v	
1,222 8	1,233.7	1,244.4	1,254.9	1,265.2	1,275.5	1,326.1	1,376.9	1,428.3	1,480.0	1,532.5	h	126
1.6221	1.6346	1.6466	1.6581	1.6692	1.6801	1.7302	1.7760	1.8185	1.8580	1.8953	s	(344.94)
3.465	3.566	3.664	3.761	3.856	3.951	4.410	4.857	5.297	5.728	6.161	v	
1,220.5	1,231.7	1,242.5	1,253.2	1,263.7	1,274.1	1,325.1	1,376.2	1,427.8	1,479.7	1,532.2	h	140
1.6084	1.6212	1.6334	1.6451	1.6564	1.6674	1.7179	1.7640	1.8066	1.8462	1.8836	s	(353.03)
3.219	3.315	3.408	3.500	3.589	3.679	4.109	4.528	4.939	5.342	5.747	v	
1,218 8	1,230.2	1,241.2	1,252.0	1,262.6	1,273.1	1,324.4	1,375.7	1,427.4	1,479.4	1,532.0	h	150
1.5993	1.6123	1.6247	1.6365	1.6479	1.6589	1.7098	1.7561	1.7988	1.8385	1.8758	s	(358.43)
2.926	3.015	3.102	3.186	3.269	3.352	3.750	4.135	4.512	4.882	5.252	v	
1,216.4	1,228.1	1,239.3	1,250.3	1,261.0	1,271.6	1,323.4	1,375.0	1,426.9	1,479.0	1,531.7	h	164
1.5872	1.6006	1.6132	1.6253	1.6368	1.6480	1.6993	1.7458	1.7887	1.8285	1.8659	s	(365.52)
2.712	2.796	2.878	2.958	3.036	3.114	3.487	3.847	4.200	4.546	4.891	v	
1,214.2	1,226.2	1,237.7	1,248.8	1,259.7	1,270.4	1,322.5	1,374.4	1,426.5	1,478.7	1,531.5	h	176
1.5774	1.5911	1.6040	1.6163	1.6279	1.6392	1.6909	1.7377	1.7808	1.8206	1.8580	s	(371.24)
2.495	2.575	2.652	2.728	2.801	2.873	3.222	3.558	3.886	4.208	4.527	v	
1,211.7	1,224.0	1,235.7	1,247.0	1,258.1	1,268.9	1,321.5	1,373.7	1,426.0	1,478.3	1,531.2	h	190
1.5666	1.5806	1.5938	1.6063	1.6182	1.6296	1.6818	1.7288	1.7721	1.8120	1.8495	s	(377.55)
2.358	2.436	2.510	2.583	2.654	2.722	3.056	3.376	3.688	3.995	4.299	v	
1,209.8	1,222.3	1,234.2	1,245.7	1,256.9	1,267.9	1,320.8	1,373.1	1,425.6	1,478.0	1,531.0	h	200
1.5592	1.5735	1.5869	1.5996	1.6116	1.6231	1.6756	1.7228	1.7662	1.8062	1.8438	s	(381.82)
2.189	2.262	2.333	2.402	2.470	2.535	2.849	3.150	3.443	3.731	4.014	v	
1,207.1	1,220.0	1,232.2	1,243.9	1,255.3	1,266.4	1,319.8	1,372.4	1,425.1	1,477.6	1,530.7	h	214
1.5492	1.5640	1.5777	1.5906	1.6028	1.6145	1.6674	1.7149	1.7585	1.7986	1.8362	s	(387.53)

"Heat Engines."

TABLE 3.—SUPERHEATED

Abs press. lb sq in (Sat temp)		Sat. water	Sat steam	Temperature— 400°	420°	440°	460°	480°	500°	520°	540°	560°
	v	0 019	2 031	2 060	2 131	2 199	2 266	2 330	2 392	2 454	2 514	2 575
226	h	366 5	1,199 3	1,204 7	1,218 0	1,230 4	1,242 3	1,253 8	1,265 1	1,276 1	1,286 9	1,297 7
(392 19)	s	0 5570	1 5347	1 5409	1 5561	1 5702	1 5832	1 5957	1 6074	1 6188	1 6297	1 6403
	v	0 0186	1 9156	1 9250	1 9938	2 060	2 124	2 185	2 244	2 303	2 360	2 418
240	h	372 1	1,200 1	1,201 9	1 215 5	1,228 3	1,240 4	1,252 1	1,263 5	1,274 7	1,285 6	1,296 5
(397 40)	s	0 5635	1 5295	1 5317	1 5473	1 5617	1 5750	1 5876	1 5996	1.6111	1 6222	1 6328
	v	0 0187	1 8410		1 9053	1 9698	2 032	2 090	2 149	2 205	2 261	2 316
250	h	376 0	1,200 5		1 213 8	1,226 7	1,239 1	1,250 9	1,262 4	1,273 7	1,284 7	1,295 6
(400 97)	s	0 5680	1 5261		1 5412	1 5558	1 5693	1 5854	1 5944	1 6058	1 6170	1 6277
	v	0 0187	1 7397		1 7846	1 8465	1 9060	1 9625	2 018	2 072	2 126	2 178
265	h	381 6	1 201 2		1 211 0	1,224 3	1,236 9	1,249 0	1,2[illegible]9 8	1 272 1	1 283 3	1,294 3
(406 12)	s	0 5744	1 5210		1 5324	1 5472	1 5611	1 5741	1 5864	1 5982	1 6095	1 6203
	v	0 0188	1 6781		1 7113	1 7718	1 8299	1 8850	1 9393	1 9920	2 044	2 094
275	h	385 2	1 201 0		1 209 2	1 222 7	1,235 5	1,247 8	1,259 6	1,271 1	1,282 4	1,293 4
(409 44)	s	0 5785	1 5178		1 5266	1 5417	1 5558	1 5690	1 5814	1 5933	1 6046	1 6156
	v	0 0189	1 5934		1 6108	1 6693	1 7257	1 7789	1 8310	1 8817	1 9313	1 9798
290	h	390 5	1 202 1		1 206 3	1 220 3	1,233 4	1,245 8	1,257 9	1,269 5	1,280 9	1,292 1
(414.24)	s	0 5845	1 5132		1 5181	1 5337	1 5480	1 5615	1 5742	1 5862	1 5977	1 6088
	v	0 0189	1 5414		1 5493	1 6067	1 6618	1 7140	1 7648	1 8143	1 8626	1 9098
300	h	393 9	1 202 4		1 204 4	1 218 6	1,231 9	1,244 6	1,256 7	1,268 5	1,280 0	1 291 2
(417 33)	s	0 5883	1 5102		1 5126	1 5284	1 5430	1 5567	1 5695	1 5816	1 5932	1 6044
	v	0 0190	1 4251			1 4664	1 5186	1 5684	1 6167	1 6636	1 7090	1 7531
325	h	402 1	1 203 1			1 214 3	1,228 1	1 241 2	1 253 8	1,265 8	1 277 5	1,289 0
(424 73)	s	0 5975	1 5032			1 5158	1 5309	1 5451	1 5583	1 5707	1 5825	1 5939
	v	0 0191	1 3245			1 3461	1 3960	1 4436	1 4899	1 5344	1 5772	1 6189
350	h	409 8	1 203 6			1 209 8	1,224 2	1,237 8	1,250 7	1,263 1	1,275 0	1,286 7
(431 71)	s	0 6061	1 4966			1 5036	1 5194	1 5340	1 5477	1 5604	1 5724	1 5840
	v	0 0192	1 2370			1 2413	1 2892	1 3353	1 3794	1 4219	1 4628	1 5025
375	h	417 2	1 203 9			1 205 2	1 220 2	1,234 2	1,247 6	1,260 3	1,272 5	1 284 4
(438 31)	s	0 6142	1 4903			1 4918	1 5083	1 5234	1 5375	1 5506	1 5629	1 5747
	v	0 0194	1 1601				1 1958	1 2403	1 2828	1 3235	1 3626	1 4007
400	h	424 2	1 204 1				1 216 0	1,230 6	1,244 3	1,257 4	1,269 9	1,282 1
(444 58)	s	0 6218	1 4843				1 4974	1 5132	1 5276	1 5411	1 5538	1 5658
	v	0 0195	1 0916				1 1122	1 1555	1 1968	1 2362	1 2739	1 3104
425	h	430 9	1,204 1				1 211 6	1,226 7	1,241 0	1,254 5	1,267 3	1,279 7
(450 56)	s	0 6292	1 4786				1 4870	1 5032	1.5182	1 5321	1 5451	1 5573
	v	0 0196	1 0303				1 0380	1 0802	1 1204	1 1586	1 1950	1 2301
450	h	437 4	1 204 1				1 207 1	1,222 8	1,237 6	1,251 5	1,264 6	1,277 3
(456 27)	s	0 6361	1 4732				1 4767	1 4936	1 5091	1 5234	1 5367	1 5493
	v	0 0197	0 9756					1 0127	1 0521	1 0891	1 1244	1 1583
475	h	443 6	1 203 9					1 218 8	1 234 1	1,248 4	1,261 9	1,274 8
(461 74)	s	0 6428	1 4680					1 4841	1 5002	1.5149	1 5286	1 5414
	v	0 0198	0 9261					0 9520	0 9905	1 0267	1 0608	1 0937
500	h	449 7	1,203 7					1 214 6	1,230 5	1,245 2	1 259 1	1,272 3
(466 99)	s	0.6493	1 4630					1 4748	1 4915	1 5067	1 5208	1 5338
	v	0 0200	0 8402					0 8449	0 8823	0 9173	0 9500	0 9812
550	h	461 3	1,203 0					1 205 7	1,222 8	1,238 6	1,253 3	1,267 1
(476 92)	s	0.6616	1 4536					1 4564	1 4747	1.4909	1 5057	1 5194
	v	0 0202	0 7677						0 7922	0 8262	0 8577	0 8876
600	h	472 3	1 202 1						1,214 7	1,231 7	1,247 2	1,261 8
(486 17)	s	0 6731	1 4447						1 4582	1 4755	1 4913	1 5056
	v	0 0204	0 7060						0 7140	0 7475	0 7786	0 8075
650	h	482 9	1,201 0						1,206 1	1,224 2	1,240 8	1,256 2
(494 86)	s	0 6840	1 4364					. .	1 4418	1 4605	1 4773	1 4925

[1] ALLEN and BURSLEY,

STEAM.[1]—(*Continued*)

degrees Fahrenheit												
580°	600°	620°	640°	660°	680°	700°	740°	800°	900°	1000°		Abs press. lb-sq in. (Sat. temp.)
2.634	2.692	2.751	2.808	2.865	2.922	2.979	3.091	3.257	3.530	3.798	v	
1,308.3	1,318.9	1,329.4	1,340.0	1,350.6	1,361.2	1,371.8	1,392.9	1,424.6	1,477.2	1,530.4	h	225
1.6507	1.6608	1.6706	1.6803	1.6898	1.6992	1.7085	1.7264	1.7522	1.7924	1.8301	s	(392.19)
2.474	2.529	2.585	2.639	2.693	2.747	2.800	2.907	3.063	3.321	3.574	v	
1,307.2	1,317.8	1,328.5	1,339.1	1,349.7	1,360.4	1,371.0	1,392.3	1,424.1	1,476.8	1,530.2	h	240
1.6433	1.6534	1.6633	1.6731	1.6827	1.6921	1.7014	1.7194	1.7453	1.7856	1.8234	s	(397.40)
2.371	2.424	2.478	2.530	2.582	2.634	2.685	2.788	2.938	3.186	3.429	v	
1,306.4	1,317.1	1,327.8	1,338.5	1,349.1	1,359.8	1,370.5	1,391.8	1,423.7	1,476.6	1,530.0	h	250
1.6382	1.6483	1.6584	1.6682	1.6778	1.6873	1.6966	1.7146	1.7406	1.7810	1.8188	s	(400.97)
2.230	2.281	2.331	2.382	2.431	2.480	2.528	2.626	2.769	3.003	3.232	v	
1,305 1	1,316.0	1,326.7	1,337.5	1,348.2	1,359.0	1,369.7	1,391.1	1,423.1	1,476.1	1,529.6	h	265
1.6309	1.6411	1.6512	1.6611	1.6708	1.6803	1.6896	1.7078	1.7338	1.7743	1.8122	s	(406.12)
2.145	2.194	2.243	2.291	2.339	2.387	2.434	2.528	2.666	2.892	3.113	v	
1,304.3	1,315.2	1,326.0	1,336.8	1,347.6	1,358.4	1,369.2	1,390.7	1,422.7	1,475.8	1,529.4	h	275
1.6262	1.6365	1.6466	1.6566	1.6663	1.6758	1.6852	1.7034	1.7295	1.7701	1.8081	s	(409.44)
2.028	2.075	2.122	2.168	2.213	2.259	2.303	2.393	2.524	2.739	2.949	v	
1,303.1	1,314.1	1,325.0	1,335.8	1,346.7	1,357.5	1,368.4	1,390.0	1,422.1	1,475.4	1,529.1	h	290
1.6195	1.6298	1.6401	1.6500	1.6598	1.6695	1.6788	1.6971	1.7233	1.7640	1.8021	s	(414.24)
1.9562	2.002	2.047	2.093	2.137	2.180	2.224	2.310	2.438	2.646	2.849	v	
1,302.3	1,313.3	1,324.2	1,335.2	1,346.0	1,356.9	1,367.8	1,389.5	1,421.7	1,475.1	1,528.9	h	300
1.6152	1.6256	1.6358	1.6458	1.6557	1.6654	1.6747	1.6931	1.7193	1.7601	1.7983	s	(417.33)
1.7968	1.8395	1.8819	1.9240	1.9652	2.006	2.046	2.127	2.245	2.438	2.626	v	
1,300 2	1,311.4	1,322.5	1,333.5	1,344.5	1,355.5	1,366.5	1,388.3	1,420.7	1,474.4	1,528.4	h	325
1.6048	1.6154	1.6258	1.6359	1.6458	1.6556	1.6651	1.6836	1.7100	1.7510	1.7893	s	(424.73)
1.6601	1.7003	1.7401	1.7795	1.8183	1.8566	1.8945	1.9691	2.080	2.260	2.435	v	
1,298 1	1,309.4	1,320.6	1,331.8	1,342.9	1,354.0	1,365.1	1,387.1	1,419.8	1,473.6	1,527.9	h	350
1.5951	1.6059	1.6164	1.6266	1.6366	1.6464	1.6561	1.6747	1.7012	1.7424	1.7809	s	(431.71)
1.5415	1.5796	1.6172	1.6542	1.6908	1.7270	1.7627	1.8331	1.9371	2.106	2.270	v	
1,296 0	1,307.5	1,318.8	1,330.1	1,341.3	1,352.5	1,363.7	1,385.9	1,418.7	1,472.9	1,527.4	h	375
1.5860	1.5969	1.6075	1.6178	1.6279	1.6379	1.6476	1.6664	1.6931	1.7344	1.7731	s	(438.31)
1.4376	1.4740	1.5096	1.5444	1.5792	1.6134	1.6472	1.7140	1.8119	1.9704	2.125	v	
1,293 9	1,305.5	1,317.0	1,328.4	1,339.7	1,351.0	1,362.3	1,384.7	1,417.7	1,472.1	1,526.8	h	400
1.5773	1.5884	1.5991	1.6095	1.6197	1.6298	1.6396	1.6585	1.6854	1.7270	1.7658	s	(444.58)
1.3456	1.3804	1.4144	1.4476	1.4808	1.5132	1.5454	1.6087	1.7013	1.8513	1.9970	v	
1,291.7	1,303.5	1,315.1	1,326.7	1,338.1	1,349.5	1,360.9	1,383.4	1,416.7	1,471.4	1,526.3	h	425
1.5690	1.5802	1.5911	1.6017	1.6120	1.6221	1.6320	1.6511	1.6782	1.7199	1.7589	s	(450.56)
1.2640	1.2972	1.3299	1.3616	1.3932	1.4242	1.4548	1.5151	1.6032	1.7455	1.8834	v	
1,289 5	1,301.5	1,313.2	1,324.9	1,336.5	1,348.0	1,359.4	1,382.1	1,415.7	1,470.6	1,525.8	h	450
1.5611	1.5725	1.5835	1.5942	1.6046	1.6148	1.6248	1.6441	1.6714	1.7133	1.7524	s	(456.27)
1.1909	1.2228	1.2542	1.2846	1.3149	1.3445	1.3738	1.4314	1.5155	1.6507	1.7817	v	
1,287.2	1,209.4	1,311.3	1,323.1	1,334.8	1,346.5	1,358.0	1,380.9	1,414.6	1,469.9	1,525.3	h	475
1.5535	1.5650	1.5762	1.5871	1.5975	1.6079	1.6179	1.6373	1.6648	1.7069	1.7462	s	(461.74)
1.1251	1.1558	1.1861	1.2153	1.2444	1.2727	1.3009	1.3561	1.4365	1.5655	1.6903	v	
1,285.0	1,297.3	1,309.4	1,321.4	1,333.2	1,344.9	1,356.6	1,379.6	1,413.6	1,469.1	1,524.8	h	500
1.5462	1.5579	1.5692	1.5802	1.5908	1.6012	1.6113	1.6309	1.6586	1.7009	1.7404	s	(466.99)
1.0110	1.0398	1.0679	1.0953	1.1223	1.1487	1.1746	1.2256	1.2996	1.4181	1.5321	v	
1,280.3	1,293.1	1,305.5	1,317.7	1,329.8	1,341.8	1,353.6	1,377.0	1,411.5	1,467.6	1,523.8	h	550
1.5322	1.5443	1.5559	1.5671	1.5780	1.5886	1.5990	1.6188	1.6468	1.6897	1.7296	s	(476.92)
0.9159	0.9431	0.9695	0.9954	1.0206	1.0452	1.0694	1.1169	1.1855	1.2953	1.4003	v	
1,275.5	1,288.7	1,301.5	1,314.1	1,326.4	1,338.6	1,350.6	1,374.4	1,409.3	1,466.1	1,522.8	h	600
1.5191	1.5316	1.5436	1.5551	1.5662	1.5770	1.5874	1.6076	1.6360	1.6794	1.7196	s	(486.17)
0.8349	0.8609	0.8860	0.9104	0.9342	0.9575	0.9802	1.0247	1.0890	1.1917	1.2886	v	
1,270.6	1,284.3	1,297.5	1,310.3	1,322.9	1,335.3	1,347.6	1,371.8	1,407.1	1,464.6	1,521.8	h	650
1.5066	1.5195	1.5319	1.5437	1.5550	1.5658	1.5766	1.5972	1.6259	1.6698	1.7108	s	(494.86)

"Heat Engines."

TABLE 3 —SUPERHEATED

Abs press. lb/sq in. (Sat. temp.)		Sat. water	Sat. steam	Temperature—								
				520°	540°	560°	580°	600°	620°	640°	660°	680°
	v	0 0206	0 6527	0 6801	0 7107	0 7389	0 7655	0 7905	0 8143	0 8376	0 8602	0 8822
700	h	492 9	1,199 7	1,216 3	1,234 1	1,250 4	1,265 5	1,279 7	1,293 3	1,306 5	1,319 4	1,332 0
(503 04)	s	0.6943	1.4285	1.4457	1.4637	1.4797	1.4945	1 5080	1 5208	1 5329	1 5444	1 5557
	v	0 0208	0 6063	0 6204	0 6510	0 6789	0 7047	0 7289	0 7520	0 7743	0 7958	0 8168
750	h	502 6	1,198 3	1,207 9	1,226 9	1,244 2	1,260 1	1,275 0	1,289 0	1,302 6	1,315 8	1,328 7
(510 80)	s	0.7041	1.4211	1.4311	1.4502	1.4673	1 4828	1.4970	1.5101	1.5226	1 5344	1 5459
	v	0 0209	0 5653	0 5681	0 5988	0 6263	0 6515	0 6750	0 6974	0 7189	0 7395	0 7596
800	h	511 8	1,196 7	1,198 9	1,219 3	1,237 8	1,254 6	1,270 1	1,284 6	1,298 6	1,312 1	1,325 3
(518 18)	s	0.7135	1.4139	1.4161	1.4368	1.4551	1.4714	1 4862	1.4998	1 5127	1 5248	1 5365
	v	0 0211	0 5292	....	0 5514	0 5790	0 6039	0 6271	0 6489	0 6696	0 6896	0 7089
850	h	520 8	1,195 0	.. .	1,211 3	1,231 0	1,248 7	1,265 0	1,280 1	1,294 5	1,308 3	1,321 8
(525 21)	s	0.7224	1.4071	..	1 4234	1.4430	1 4603	1 4758	1.4898	1 5031	1 5156	1 5275
	v	0 0213	0 4969		0 5094	0 5369	0 5616	0 5844	0 6057	0 6258	0 6452	0 6639
900	h	529 5	1,193 3		1,202 7	1,223 8	1,242 7	1,259 8	1,275 5	1,290 3	1,304 5	1,318 3
(531 95)	s	0.7311	1.4005		1 4100	1 4309	1.4493	1.4656	1.4803	1 4938	1 5067	1 5188
	v	0 0215	0 4679		0 4700	0 4983	0 5232	0 5458	0 5668	0 5864	0 6052	0 6234
950	h	537 9	1,191 5		1,193 5	1,216 2	1,236 3	1,254 3	1,270 7	1,286 0	1,300 6	1 314 7
(538 40)	s	0 7394	1.3942		1 3962	1.4187	1.4384	1.4555	1.4708	1.4848	1 4980	1 5104
	v	0 0217	0 4419			0 4635	0 4885	0 5111	0 5317	0 5509	0 5692	0 5870
1,000	h	546 0	1,189 6			1 208 3	1,229 7	1,248 7	1,255 8	1,281 6	1,296 6	1,311 0
(544 58)	s	0.7473	1.3881			1 4065	1 4274	1.4455	1.4615	1 4760	1 4895	1 5022
	v	0 0222	0 3960			0 4016	0 4274	0 4500	0 4705	0 4893	0 5071	0 5239
1,100	h	561 7	1,185 6			1 190 7	1,215 2	1,236 6	1,255 4	1,272 5	1,288 3	1,303 4
(556 28)	s	0.7624	1.3765			1 3813	1 4054	1 4257	1.4433	1 4590	1 4733	1 4866
	v	0 0226	0 3582				0 3752	0 3985	0 4189	0 4373	0 4547	0 4710
1,200	h	576 5	1,181 4				1,199 1	1,223 4	1,244 2	1,262 7	1,279 6	1,295 5
(567 14)	s	0 7765	1.3656				1.3828	1.4058	1 4254	1.4423	1 4576	1 4716
	v	0 0230	0 3259				0 3297	0 3537	0 3746	0 3932	0 4100	0 4260
1,300	h	590 6	1,177 0				1 181 0	1 208 7	1,232 1	1,252 4	1,270 5	1,287 2
(577 32)	s	0.7897	1.3552				1.3591	1 3855	1.4074	1.4260	1 4423	1 4571
	v	0 0235	0 2983					0 3144	0 3361	0 3548	0 3717	0 3874
1,400	h	604 3	1,172 4					1 192 4	1,218 9	1,241 3	1,260 8	1,278 6
(586 96)	s	0.8024	1.3452					1 3643	1 3891	1.4096	1 4272	1 4429
	v	0 0239	0 2741					0 2789	0 3017	0 3211	0 3381	0 3536
1,500	h	617 5	1,167 6					1 174 3	1 204 3	1,229 3	1,250 5	1,269 6
(596 08)	s	0.8146	1.3357	.				1 3420	1 3701	1.3930	1 4122	1 4290
	v	0 0249	0 2338	..					0 2422	0 2636	0 2817	0 2975
1,700	h	642 5	1,157 5						1,170 4	1,202 0	1,227 9	1,250 1
(612 98)	s	0.8373	1.3174						1.3294	1 3584	1 3818	1 4014
	v	0 0265	0 1875							0 1931	0 2145	0 2320
2,000	h	679 0	1,139 0							1 149 0	1,186 6	1,216 0
(635 61)	s	0.8696	1.2896							1 2988	1 3326	1 3586
	v	0 0284	0 1510							...	0 1571	0 1796
2,300	h	716 4	1,115 2								1,128 1	1,173 1
(655 70)	s	0.9021	1.2596								1 2713	1 3110
	v	0 0301	0 1303									0 1479
2,500	h	742 8	1,095 6	.								1 135 0
(667 98)	s	0.9247	1.2375									1 2723
	v	0 0367	0 0844									
3,000	h	823 1	1,025 6	.								
(695 25)	s	0.9922	1.1676	.								
	v	0 0522	0 0522									
3,226	h	925 0	925 0									...
(706.10)	s	1 0785	1 0785		...	......	.	.....	...	....	.....	....

[1] ALLEN and BURSLEY,

STEAM.[1]—(*Continued*)

degrees Fahrenheit										
700°	**720°**	**740°**	**760°**	**780°**	**800°**	**850°**	**900°**	**950°**	**1000°**	**Abs press. lb/sq in. (Sat. temp.)**
0.9038	0.9250	0.9457	0.9661	0.9864	1.0063	1.0546	1.1029	1.1479	1.1929	*v*
1,344.5	1,356.9	1,369.1	1,381.1	1,393.1	1,404.9	1,434.0	1,460.3	1,491.9	1,520.8	*h* **700**
1.5665	1.5771	1.5873	1.5973	1.6070	1.6165	1.6387	1.6608	1.6813	1.7018	*s* **(503.04)**
0.8373	0.8574	0.8772	0.8966	0.9158	0.9348	0.9808	1.0254	1.0682	1.1100	*v*
1,341.4	1,353.9	1,366.3	1,378.5	1,390.7	1,402.7	1,432.2	1,461.5	1,490 6	1,519.8	*h* **750**
1.5569	1.5676	1.5780	1.5882	1.5980	1.6076	1.6306	1.6524	1.6735	1.6938	*s* **(510.80)**
0.7791	0.7983	0.8172	0.8358	0.8541	0.8723	0.9157	0.9577	0.9982	1.0374	*v*
1,338.2	1,350.9	1,363.5	1,375.9	1,388.2	1,400.4	1,430.3	1,459 9	1,489.3	1,518.8	*h* **800**
1.5477	1.5586	1.5692	1.5795	1.5894	1.5992	1.6225	1.6446	1.6659	1.6864	*s* **(518.18)**
0.7278	0.7462	0.7642	0.7821	0.7996	0.8168	0.8582	0.8981	0.9364	0.9735	*v*
1,335.0	1,347.9	1,360.7	1,373.3	1,385.7	1,398.1	1,428.4	1,458.3	1,488.1	1,517.8	*h* **850**
1.5389	1.5500	1.5607	1.5712	1.5812	1.5911	1.6148	1.6372	1.6587	1.6794	*s* **(525.21)**
0.6821	0.6999	0.7171	0.7344	0.7511	0.7675	0.8072	0.8451	0.8815	0.9166	*v*
1,331.7	1,344.8	1,357.8	1,370.6	1,383.2	1,395.8	1,426.5	1,456.8	1,486.8	1,516.8	*h* **900**
1.5304	1.5417	1.5526	1.5632	1.5734	1.5835	1.6074	1.6301	1.6518	1.6727	*s* **(531.95)**
0.6410	0.6582	0.6749	0.6915	0.7076	0.7234	0.7614	0.7975	0.8321	0.8657	*v*
1,328.3	1,341.7	1,354.8	1,367.8	1,380.7	1,393.4	1,424.6	1,455.2	1,485.6	1,515.8	*h* **950**
1.5223	1.5337	1.5448	1.5555	1.5660	1.5761	1.6004	1.6234	1.6452	1.6663	*s* **(538.40)**
0.6040	0.6207	0.6369	0.6529	0.6684	0.6837	0.7202	0.7547	0.7877	0.8199	*v*
1,324.9	1,338.5	1,351.9	1,365.1	1,378.1	1,391.0	1,422.6	1,453.6	1,484.3	1,514.8	*h* **1,000**
1.5144	1.5260	1.5373	1.5481	1.5588	1.5691	1.5936	1.6169	1.6390	1.6603	*s* **(544.58)**
0.5401	0.5559	0.5712	0.5863	0.6009	0.6152	0.6490	0.6810	0.7115	0.7408	*v*
1,317.9	1,332.0	1,345.8	1,359.4	1,372.8	1,386.1	1,418.7	1,450.4	1,481.8	1,512.8	*h* **1,100**
1.4993	1.5113	1.5229	1.5341	1.5450	1.5557	1.5810	1.6048	1.6274	1.6491	*s* **(556.28)**
0.4865	0.5015	0.5162	0.5304	0.5444	0.5578	0.5897	0.6195	0.6478	0.6750	*v*
1,310.6	1,325.2	1,339.5	1,353.5	1,367.4	1,381.1	1,414.7	1,447.2	1,479.3	1,510 8	*h* **1,200**
1.4848	1.4972	1.5092	1.5208	1.5321	1.5431	1.5692	1.5937	1.6168	1.6388	*s* **(567.14)**
0.4412	0.4557	0.4699	0.4834	0.4966	0.5095	0.5394	0.5675	0.5940	0.6190	*v*
1,303 0	1,318.2	1,333.0	1,347.5	1,361.8	1,376.0	1,410.6	1,444.0	1,476.8	1,508.8	*h* **1,300**
1.4709	1.4839	1.4962	1.5082	1.5199	1.5312	1.5581	1.5832	1.6068	1.6292	*s* **(577.32)**
0.4021	0.4163	0.4299	0.4430	0.4556	0.4678	0.4963	0.5229	0.5476	0.5712	*v*
1,295.2	1,311.0	1,326.3	1,341.3	1,356.1	1,370.8	1,406.5	1,440.8	1,474.3	1,506.9	*h* **1,400**
1.4574	1.4709	1.4837	1.4961	1.5082	1.5199	1.5476	1.5736	1.5976	1.6204	*s* **(586.96)**
0.3682	0.3819	0.3951	0.4077	0.4200	0.4318	0.4590	0.4840	0.5076	0.5298	*v*
1,287.1	1,303.6	1,319.4	1,334.9	1,350.2	1,365.4	1,402.3	1,437.6	1,471.8	1,504.9	*h* **1,500**
1.4442	1.4583	1.4716	1.4844	1.4969	1.5091	1.5377	1.5642	1.5889	1.6120	*s* **(596.08)**
0.3119	0.3250	0.3376	0.3496	0.3612	0.3721	0.3974	0.4204	0.4416	0.4616	*v*
1,269.8	1,288.0	1,305.1	1,321.7	1,338.0	1,354.4	1,393.7	1,431.1	1,466.8	1,500.8	*h* **1,700**
1.4185	1.4340	1.4484	1.4621	1.4755	1.4885	1.5191	1.5472	1.5730	1.5968	*s* **(612.98)**
0.2468	0.2601	0.2723	0.2837	0.2945	0.3047	0.3279	0.3486	0.3673	0.3847	*v*
1,240.7	1,262.4	1,282.1	1,300.6	1,318.9	1,337.0	1,380.4	1,421.1	1,459.2	1,494.7	*h* **2,000**
1.3802	1.3987	1.4152	1.4306	1.4454	1.4599	1.4937	1.5242	1.5518	1.5765	*s* **(635.61)**
0.1963	0.2105	0.2230	0.2343	0.2446	0.2543	0.2761	0.2954	0.3123	0.3279	*v*
1,206.1	1,233.3	1,256.8	1,278.1	1,298.5	1,318.6	1,366.4	1,410.7	1,451.3	1,488.2	*h* **2,300**
0.3397	1.3630	1.3828	1.4004	1.4170	1.4331	1.4702	1.5035	1.5329	1.5586	*s* **(655.70)**
0.1676	0.1829	0.1960	0.2074	0.2177	0.2272	0.2484	0.2668	0.2829	0.2976	*v*
1,178.4	1,211.0	1,238.1	1,261.9	1,284.2	1,305.8	1,356.7	1,403.5	1,446.0	1,483.9	*h* **2,500**
1.3101	1.3379	1.3608	1.3804	1.3985	1.4158	1.4555	1.4905	1.5213	1.5477	*s* **(667.98)**
0.0983	0.1246	0.1409	0.1538	0.1646	0.1742	0.1947	0.2118	0.2264	0.2396	*v*
1,066.3	1,138.3	1,181.7	1,215.5	1,244.5	1,271.1	1,331.0	1,384.3	1,431.6	1,472.9	*h* **3,000**
1.2028	1.2644	1.3008	1.3288	1.3524	1.3737	1.4203	1.4602	1.4944	1.5233	*s* **(695.25)**
....	0.0993	0.1195	0.1338	0.1453	0.1552	0.1757	0.1924	0.2066	0.2192	*v*
....	1,088.7	1,149.1	1,190.6	1,224.2	1,253.8	1,318.7	1,375.0	1,424.7	1,467.8	*h* **3,226**
....	1.2183	1.2691	1.3034	1.3307	1.3545	1.4050	1.4472	1.4831	1.5133	*s* **(706.10)**

TABLE 4.—MOLECULAR AND EQUIVALENT WEIGHTS OF ELEMENTS, COMPOUNDS, AND RADICALS

Substance	Formula	Weight of molecules or radicals	Equivalent weight	Effect in boiler*	Saturated solution grains per U S gallon	
					At 60°F	At 210°F
Aluminum	Al	27 1	9 03			
Sodium aluminate	$Na_2Al_2O_4$	164 2	27 36			
Barium	Ba	137 4	68 70			
Barium sulphate	$BaSO_4$	233 5	116 75			
Calcium	Ca	40 0	20 00			
Calcium bicarbonate	$Ca(HCO_3)_2$	162 0	81 00	S		
Calcium carbonate	$CaCO_3$	100 0	50 00	S	2 5	1 5
Calcium chloride	$CaCl_2$	110 9	55 45	S C	21 650	35 700
Calcium hydrate (pure)	$Ca(OH)_2$	74 0	37 00			
Calcium oxide (pure)	CaO	56 0	28 00			
Calcium sulphate (anhydrous)	$CaSO_4$	136 0	68 00	S C	140	125
Calcium nitrate	$Ca(NO_3)$	164 1	82 05	S C		
Iron (ferrous)	Fe	56 0	28 00			
Iron (ferric)	Fe '	56 0	18 66			
Magnesium	Mg	24 4	12 20			
Magnesium bicarbonate	$Mg(HCO_3)_2$	146 4	73 20	S		
Magnesium carbonate	$MgCO_3$	84 4	42 20	S	1 0	1 8
Magnesium chloride	$MgCl_2$	95 3	47 65	S C	97 800	216 000
Magnesium hydrate	$Mg(OH)_2$	58 4	29 20			
Magnesium nitrate	$Mg(NO_3)_2$	148 4	74 20			
Magnesium sulphate	$MgSO_4$	120 4	60 20	S C	45 000	392 500
Potassium	K	39 1	39 10			
Silica	SiO_2	60 4	30 20	S		
Sodium	Na	23 05	23 05			
Sodium bicarbonate	$NaHCO_3$	84 6	84 60	F E	4 100	9 380
Sodium carbonate	Na_2CO_3	106 0	53 00	F E		
Sodium chloride	NaCl	58 5	58 50	F C	22 200	23 400
Sodium hydrate	NaOH	40 1	40 10			
Sodium nitrate	$NaNO_3$	85 0	85 00	F E		
Tri-sodium phosphate	$Na_3PO_4\ 12H_2O$	380 4	126 90			
Di-sodium phosphate	$Na_2HPO_4\ 12H_2O$	358 2	119 46			
Mono-sodium phosphate	$NaH_2PO_4\ H_2O$	138 1	46 03			
Sodium sulphate	Na_2SO_4	142 0	71 00	F C	7 040	182 500
Acid Radicals						
Bicarbonate	HCO_3	61 0	61 00			
Carbonate	CO_3	60 0	30 00			
Carbon dioxide	CO_2	44 0	22 00			
Chloride	Cl	35 45	35 45			
Nitrate	NO_3	62 0	62 00			
Hydrate	OH	17 0	17 00			
Phosphate	PO_4	95 0	31 66			
Sulphate	SO_4	96 06	48 03			
Acids						
Hydrogen	H	1 0	1 00			
Carbonic acid	H_2CO_3	62 0	31 00	C		
Hydrochloric acid	HCl	36 45	36 45	C		
Phosphoric acid	H_3PO_4	98 0	32 66			
Sulphuric acid	H_2SO_4	98 1	49 05	C		

NOTE —The equivalent weight of any substance is that weight which takes the place of one atom of hydrogen in compounds

* Symbols C = corrosion F = foaming and priming E = embrittlement; S = scale

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